15 May 2000

ADMA-OPCO

On-site Training Course

Production / Process

Module - 4

Pumps

Gap Elimination Program

Production / Process

Module - 4

PUMPS

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OBJECTIVES

Upon the completion of this module, the developee will be able to:

⇒ Explain what is the Net Positive Suction Head of a pump.

⇒ Calculate the Horsepower of a pump.

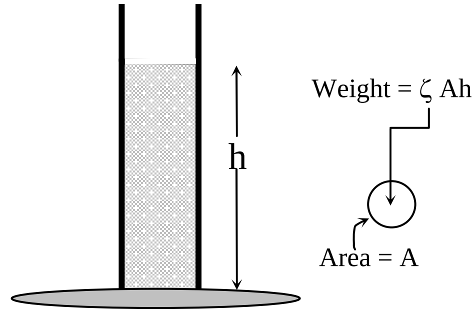
⇒ Determine pump performance and how to use pump performance curves.

⇒ Explain pump classifications.

⇒ Discuss troubleshooting of centrifugal and reciprocating pumps.

1. **Hydrodynamics**

Head



$$\text{Weight} = \text{Density x Area x height}$$

$$= \zeta \times A \times h$$

$$\text{Pressure (P)} = \frac{\text{Weight}}{\text{Area}}$$

$$= \frac{\zeta \times A \times h}{A} = \zeta h \ \text{lb/ft}^2$$

$$= \frac{\zeta h}{144} \ \text{psi}$$

ζ	=	Density of fluid,	lbs/ft^3
A	=	Area	ft^2
h	=	height of fluid,	ft

$$\text{Specific Gravity} = \gamma = \frac{\zeta \text{ fluid}}{\zeta \text{ water}}$$

$$\zeta \text{ water} = 62.34 \ \text{lbs/ft}^3$$

$$\zeta \text{ fluid} = 62.34\,\gamma$$

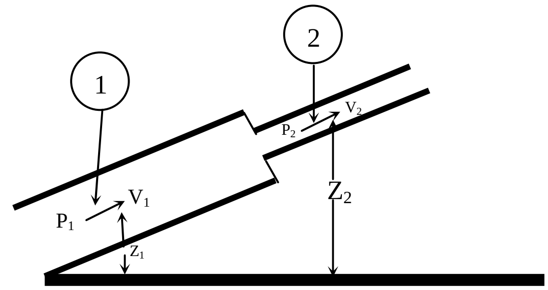
$$P = \frac{62.34\gamma h}{144}$$

$$= \frac{\gamma \times h}{2.31} = 0.433 \times \gamma \times h$$

$$\boxed{P = \frac{\gamma \times h}{2.31} = \mathbf{0.433 \times \gamma \times h}}$$

Continuity Equation

$(\zeta \times A \times V)_1 = (\zeta \times A \times V)_2$
V = velocity ft/sec
ζ = density, lb/ft^3



$A_1 \times V_1 = A_2 \times V_2$ (same liquid)

$P_1 > P_2$
$V_1 < V_2$
$Z_1 < Z_2$

Bernoulli's Equation

$$\frac{V1^2}{2g} + \frac{P_1}{\zeta} + Z_1 = \frac{V_2^2}{2g} + \frac{P_2}{\zeta} + Z_2 + h_f$$

$\frac{V^2}{2g}$ = Velocity head, ft

V = average velocity of liquid, ft/sec = 0.408 GPM/d^2
g = gravitational acceleration = 32.2 ft/sec^2

Total Dynamic Head (TDH)

It is the difference between the discharge and suction heads of the pumping system.

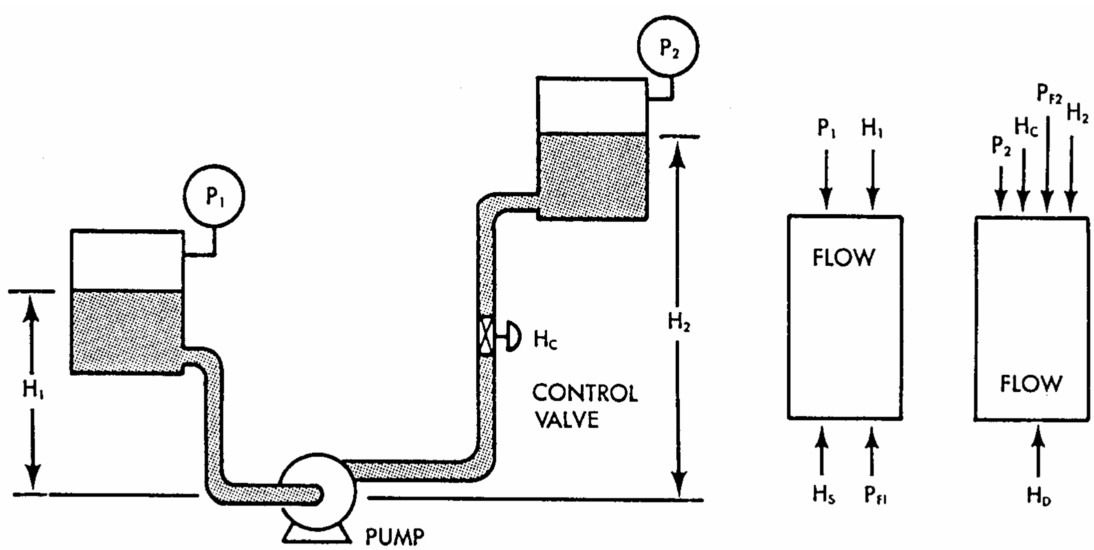
As shown in Figure the suction head is the sum of the suction vessel operating gage pressure (converted to feet), the vertical distance between the suction vessel fluid level and the pump reference point, less the head losses in the suction piping (discounting velocity head):

$$Hs = \frac{P_1 \times 2.31}{\gamma} + H_1 - \frac{(P_{f1} \times 2.31)}{\gamma}$$

$$= \frac{(Pi - Pfi)(2.31)}{\gamma} + H_1$$

where:

Hs = suction head of the liquid being pumped, ft.
P_1 = suction vessel operating pressure, psig.
H_1 = height of fluid in suction vessel above pump reference point, ft.
P_{f1} = pressure drop due to friction in the suction piping, psi.
γ = specific gravity of the liquid being pumped



The discharge head is the sum of the discharge vessel operating gage pressure (converted to feet), the liquid level in the discharge vessel above the pump reference point, pressure drop due to friction in the discharge piping, and control losses (discounting velocity head):

$$H_D = \frac{(P_2 \text{ x } 2.31)}{\gamma} + H_2 + \frac{P_{f2} \text{ x } 2.31}{\gamma} + \frac{P_c \text{ x } 2.31}{\gamma}$$

$$= \frac{(P_2 + P_{f2} + P_c)(2.3\ 1)}{\gamma} + H_2$$

where:

HD= discharge head of liquid being pumped, ft.
P_2 = discharge vessel operating pressure, psig.
H_2 = operating or 'normal height of liquid in the discharge vessel above the pump reference point, ft.
P_{f2} = pressure drop due to friction in the discharge piping, psi.
P_c = discharge flow control valve losses, ft.

The pump total dynamic head is the difference between the suction and discharge heads:

TDH = HD- Hs

$$= \frac{(P_2 - P_1 + P_{fl} + P_{f2} + P_c)(2.31)}{\gamma} + H_2 - H_1$$

where:

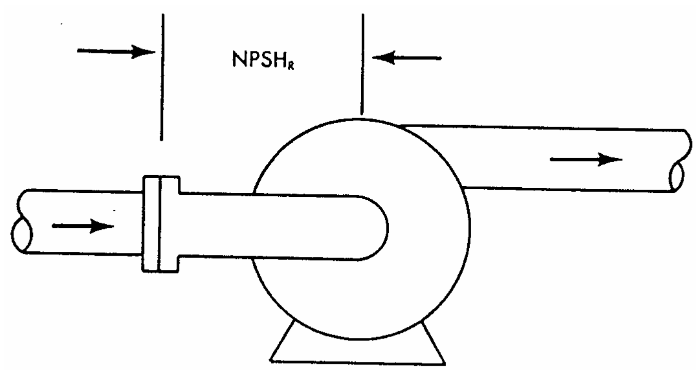
TDH = *total dynamic head* required of a pump or pumps, ft.

Note that the value of TDH varies directly with the suction and discharge vessel operating pressures and with the magnitude of the piping head and control losses; and is inversely related to the fluid specific gravity.

A pump's required discharge pressure rating is the combination of the maximum expected suction pressure and the maximum pressure developed by the pump. For reciprocating pumps, this requires the conversion to units of pressure for the previously calculated values of suction head and total dynamic head. For centrifugal pumps, the conversions are not necessary.

2. <u>Net Positive Suction Head</u>

A major problem encountered in many pumping applications, especially with liquids operating at or near their vapor pressures (bubble point), is a lack of adequate *net positive suction* head (NPSH). NPSH is the total suction head in feet of liquid *(absolute* at the pump centerline or impeller eye) less the vapor pressure (in feet) of the liquid being pumped. Based on industry convention, *net positive suction head available* ($NPSH_A$) is the amount of NPSH available at the pump suction. *Net positive suction head required* ($NPSH_R$) is the amount of NPSH required to move and accelerate the fluid from the pump suction into the pump itself (Figure 12). $NPSH_A$ must be equal to, or greater than, $NPSH_R$ in order not to cavitate or flash across the pump. The result of cavitation is decreased efficiency, capacity, and head. In severe cavitation (sounds like gravel is being pumped), the pump will experience vibration and a severely reduced life. Flashing results as the pump cavity is filled with vapors, and as a result, the pump becomes *vapor locked.* The end result of this operation is usually pump seizure.



NPSHR = HEAD LOSS IN FEET OF PUMPED FLUID FROM THE SUCTION FLANGE TO THE IMPELLER EYE OR SUCTION VALVE

Figure 12. Net Positive Suction Head Required (NPSHR)

<u>Net Positive Suction Head Required</u>

$NPSH_R$ is determined either by test or calculation by the pump manufacturer for the particular pump under consideration. For all pumps, $NPSH_R$ is a function of many variables including the fluid geometry and the smoothness of the surface areas. For centrifugal pumps, other factors that are included are the type of impeller, design of the impeller eye, and rotational speeds.

$NPSH_R$ is determined on the basis of handling cool water. Field experience and laboratory testing have confirmed that centrifugal pumps handling certain gas free hydrocarbon fluids and water at elevated temperatures will operate satisfactorily with harmless cavitation and less $NPSH_A$ than would be required for cold water. Under these conditions, the *Hydraulic Institute* allows for the reduction in the manufacturer stated $NPSH_R$ values; however, unless the piping is very well defined and the suction liquid level is very well controlled, these reductions are not generally used because the reductions are relatively small. These reductions are, however, allowed to exist as small safety factors.

Net Positive Suction Head Available

$NPSH_A$ is not a function of the pump itself, but of the piping system for the pump. It can be calculated from the following relationship (Figure 13):

$$NPSH_A = \frac{(P_1 + P_a - P_v - P_{f1})\,(2.31)}{\gamma} + H_1$$

where:

P_a = atmospheric pressure, psia.
P_v = liquid vapor pressure at pumping temp., psia.

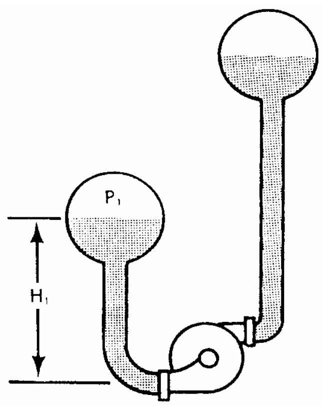


Figure 13. Net Positive Suction Head Available (NPSHA)

Figure 14 is a, vapor pressure curve for water. For other fluids, refer to the *GPSA Engineering Data Book.*

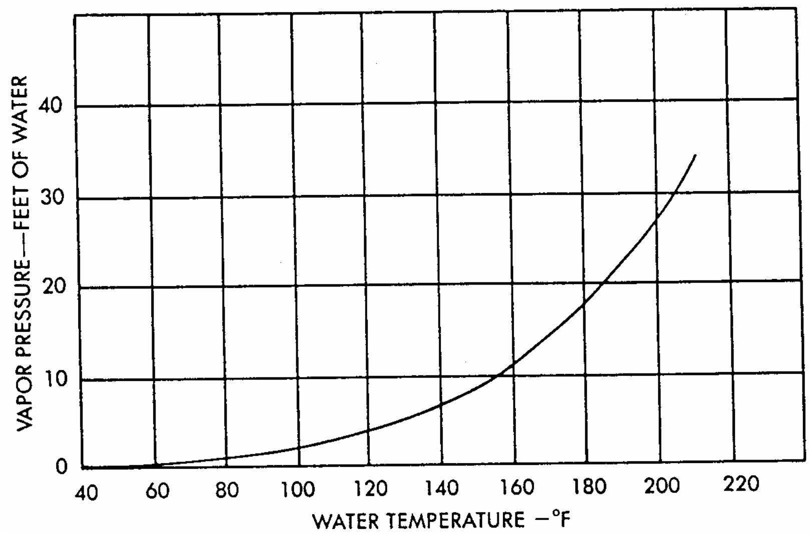


Figure 14. Water Vapor Pressure Chart

It is very important to note that $NPSH_A$ decreases linearly with increases in fluid temperatures and pipe friction losses; however, because pipe friction losses vary as the square of the flow, $NPSH_A$ also varies as the square of the flow. Therefore, $NPSH_A$ will be the lowest at the maximum flow requirement. It is then critical to recognize the need for calculating $NPSH_A$ (and $NPSH_R$) at the maximum flow conditions as well as maximum fluid temperature, not just at design.

Unless subcooled, a pure component hydrocarbon liquid is typically in equilibrium with the vapors in a pressure vessel. Therefore, increases in the vessel operating pressure are almost fully offset by a corresponding increase in the vapor pressure. When this happens,

$$NPSH_A = H_1 - P_{f1} \frac{(2.31)}{\gamma}$$

NPSH Margin

Conservatism is provided in design of NPSH by applying a safety factor; however, caution must be exercised in using too much safety factor; as it increases the purchase costs of pumps. *NPSH margin* is simply the difference between $NPSH_A$ and $NPSH_R$. Some vendors and contractors will only require that this margin be equal to, or greater than, zero. That will suffice only when the piping and fluid conditions are extremely well known. The *Hydraulic Institute* recommends that this margin be at least 3 feet, but preferably upwards of 7 feet. Perhaps a better approach is to specify a pump with a $NPSH_A$ that is less than what was calculated. For this approach, use the following relationship:

Specified $NPSH_A$ = Calculated $NPSH_A$ / Safety Factor

Typically, a safety factor of 1.0 would be used only for an existing system; a factor of 1.10 for new services which will have stable and well controlled suction conditions; and a factor of 1.25 for new or old services which tend to have rapid, frequent, or severe fluctuations in suction conditions, such as a boiler feedwater pump.

The availability of pumps and pump models depends upon the calculated value of $NPSH_A$. With $NPSH_A$ values between 1 and 7, only a small number of pumps could be selected. The ranges of commercially available pumps increase with increases in $NPSH_A$ to the point that virtually any pump can be considered if the $NPSH_A$ is above 25 feet.

Inadequate Suction Conditions

When a new system offers insufficient NPSH margin for optimum pump selection, either the $NPSH_A$ must be increased, or the $NPSH_R$ decreased, or both. To increase $NPSH_A$ the following may be considered: *raise the liquid level, lower the elevation of the selected pump, change to a low $NPSH_R$ pump, reduce suction piping friction losses, use a booster pump, or cool the liquid.* To reduce $NPSH_R$: use *different design impellers, inducers, or several smaller pumps with lower $NPSH_R$'s in parallel.* Inducers, however, have severely restricted operating flow rangeabilities relative to non-inducer pumps. Furthermore, *inducers should never be used in erosive services,*

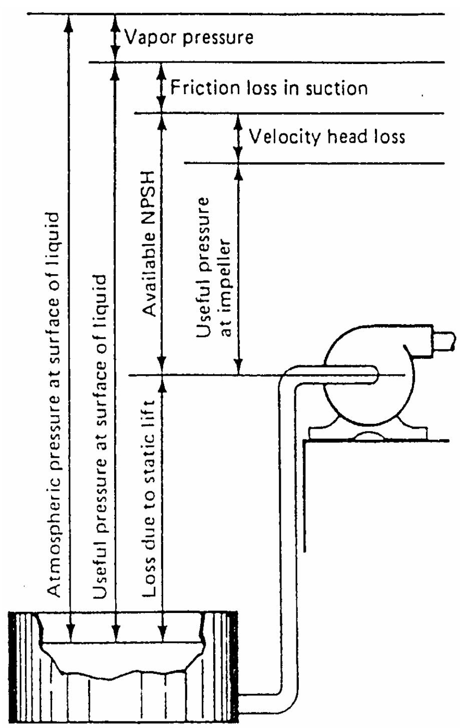
When an existing pumping system exhibits insufficient NPSH margin, it is too late to use most of the above solutions without going through an expensive change. A great majority of these type problems can be traced to suction line flow restrictions (orifice plates, plugged strainers, partially closed valves, etc.) and inadequate source tank fluid levels (including vortexing).

Total Energy vs Available NPSH

When a pump is in a suction lift condition, the only energy available is the atmospheric pressure-14,7 psia, or approximately 34 ft of water, Several factors combine to reduce the effect of this atmospheric pressure. The amount of energy remaining after the energy consumption factors have been satisfied is called available Net Positive Suction Head (NPSH).

When liquid in a column is placed under a vacuum, a portion of the liquid will evaporate and form vapor, which will reduce available energy. This vapor pressure increases with an increase in liquid temperature.

Energy is required to overcome the distance that the liquid must be lifted, to make the liquid move (velocity head), and to overcome the friction of the pipe and fittings (friction head or head loss). The available NPSH, then, is the atmospheric pressure minus vapor pressure, static suction lift, and head loss due to friction. The pressure in psi at the eye of the impeller is the NPSH minus velocity head. Figure 1.8 illustrates this concept.



AVAILABLE NPSH - SUCTION LIFT CONDITION

The largest single contributing factor to the reduction of pressure at the impeller is the lift itself. Even small changes in the height of liquid below the eye of the impeller will have a drastic effect upon he amount of liquid pumped.

From an absolute pressure standpoint, the liquid must enter the impeller eye under positive pressure in order for the pump to function. The higher the positive pressure, the greater the pump discharge.

When the eye of the impeller of a pump is below the level of the liquid source which supplies the suction (suction head condition), the available NPSH is the sum of atmospheric pressure and the height of the liquid above the eye of the impeller minus vapor pressure and minus friction loss.

Available NPSH vs Required NPSH

The NPSH discussion so far has been aimed at determining the available NPSH. Most pump curves will give a required NPSH as a part of the curve. It is necessary that the available NPSH be least as large as the required NPSH in order for the curves to be valid.

Under a suction lift condition, the NPSH should be calculated. However, for suction head conditions. the usual practice is to measure the height of liquid above the eye of the impeller. If this distance meets or exceeds the required NPSH, the pump conditions, as defined by the curve , can be used. The assumption here is that atmospheric pressure will be sufficient to overcome losses due to friction, velocity head, and vapor pressure.

3. Horsepower Calculations

In a typical pump installation, there are three related horsepower calculations that you should understand (Figure 1.7). The hydraulic power that the pump transfers to the pumped liquid is called liquid horsepower. The horsepower input to the pump is referred to as brake horsepower. This value is greater than the liquid horse power by a factor representing, the efficiency of the pump. The electrical horsepower required to run the motor (to deliver the required brake horsepower) is larger than the brake horsepower by a factor representing the efficiency of the motor.

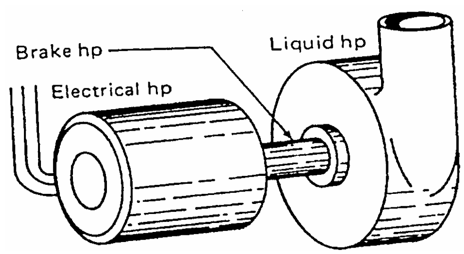


FIGURE 1.7 THREE KINDS OF HORSEPOWER

a. Liquid Horsepower

The calculations must start with the liquid horsepower requirement, The amount of energy required to raise liquid a given amount is measured in foot- pounds (ft-1b). One ft-lb of energy is the amount of energy needed to raise 1 lb of liquid 1 ft.

The energy required to move the liquid is affected by total head. The weight of the liquid moved is a reflection of the flow converted to weight in pounds. For example, moving 100 gallons of water is the same as moving 834 lb of water, since one gallon of water weighs 8.34 lb.

If you were to move 100 gallons of water through 200 ft of total head, the energy required would be:

200 ft x (100 gal x 8.34 lb/gal) = 166,800 ft-lb

If you were to move this 100 gallons in 1 minute, the energy consumption would be 166,800 ft-1b/min.

Since the common method of expressing energy consumption is horsepower, you need to convert ft- 1 b/min to horsepower, The conversion is as follows:

1 hp = 33,000 ft-1b/min

Therefor, energy consumption is

$$\frac{166{,}800 \text{ ft-1b/min}}{33{,}000 \text{ ft- lb/ min/ hp}} = 5 \text{ hp}$$

This 5 hp is liquid horsepower, the true energy transferred from the pump to the liquid. Now that all of these conversions have been made, you can write a new equation for the calculation of liquid horsepower.

$$\text{Liquid hp} = \frac{\text{Total head (ft) x flow (gpm) x 8.34 lb/gal}}{\text{33,000 ft-lb/ min/hp}}$$

b. Brake Horsepower

To compute brake horsepower, you need to know the efficiency of the pump. For example assume that the pump you were using in the previous example is 75% efficient, The brake horsepower is calculated as follows:

$$\text{Brake hp} = \frac{\text{Liquid hp}}{\text{\% Efficiency of pump}} = \frac{5 \text{ hp}}{0.75} = 6.67$$

This is the energy input required by the pump. It is also the energy output required of the motor.

c. Electrical Horsepower

The computation of electrical horsepower is based on brake horsepower and motor efficiency and is determined as follows:

$$\text{Electrical hp} = \frac{\text{Brake hp}}{\text{\% Efficiency of motor}}$$

If you assume for the above condition that the motor is 90% efficient, then the electrical horsepower is as follows:

$$\text{Electrical hp} = \frac{\text{6.67 Brake hp}}{0.90} = \text{7.4 Electrical hp.}$$

It took 5 hp to get the job done, but you had to purchase 7.4 hp. The 2.4 hp increase is lost to heat in the motor and pump. In reality, this is not considered to be an exceptionally large loss.

Power Requirements

Once pump TDH has been calculated, the power requirements can also be determined using the following formula:

$$BHp = \text{Hydraulic Hp/ Efficiency}$$

$$= \frac{(GPM)\,(TDH)\,(\gamma)}{(3960)\,(e)}; \quad \text{Kinetic Energy pumps}$$

$$= \frac{(GPM)\,(\Delta P)}{(1714)\,(e)}; \quad \text{Positive Displacement pumps}$$

where:

BHp = brake horsepower.

ΔP = maximum working pressure of positive displacement pumps, psig.

e = pump efficiency factor, obtained from the pump manufacturer or approximated from Figure 15.

For electric-motor driven pumps, the energy consumption can be estimated by:

$$\text{KWHR / day} = \frac{17.9\,(BHp)}{\text{Motor Efficiency}}$$

Note that changes in the specific gravity affects the power calculations. The pump efficiencies can be estimated from Figure 15 for either a positive displacement or centrifugal pump. Figure 15 should only be used for estimating purposes as it is far better to depend on the manufacturer stated values.

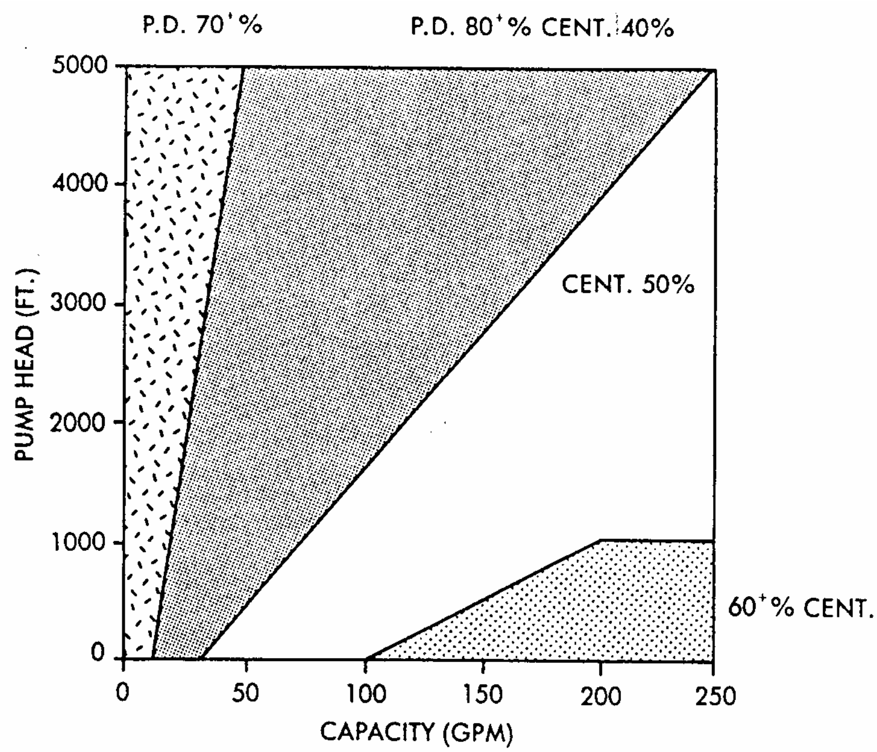


Figure 15. Pump Efficiency Chart-Screening Estimates Only

Figure 15 is also useful to determine whether a particular pumping application is best met by using a positive displacement or centrifugal pump. For high TDHs and low flow rates, a positive displacement is preferred because of its higher efficiencies. On the other hand, for high flow rates at low TDHS, it is advisable to use a centrifugal pump. For the areas of moderate TDHs and flows, the engineer could perhaps use either type of pump, depending on factors other than efficiency.

Example-1 - **Pump Head and Horsepower Calculation**

Problem : A pump takes suction from a vessel at 15 psig discharges to a tower operating at 500 psig. Using Figure 16 and given the following data, determine the following for both kinetic and positive displacement pumps:

a. pump suction head
b. pump discharge head
c. total dynamic head
d. pump horsepower required
e. electrical energy consumption

Flow	=	150 gpm
Specific gravity	=	0.8
Line losses	=	2 psi suction
	=	4 psi discharge
Control losses	=	10 psi
Static head	=	15 ft. suction
	=	80 ft. discharge
Atmospheric pressure	=	14.7 psia
Kinetic pump efficiency	=	0.52
Positive displacement pump efficiency	=	0.82
Motor efficiency	=	0.92

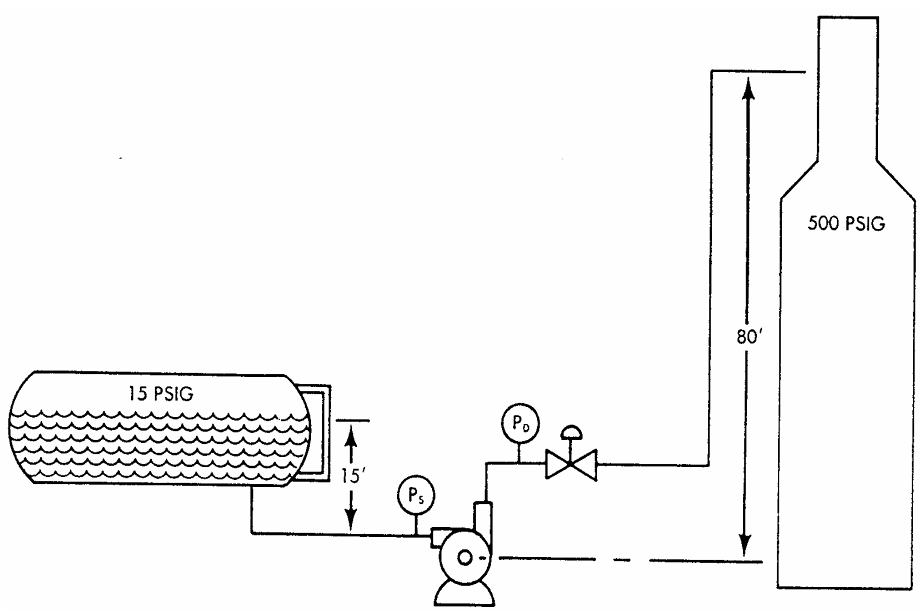


Figure 16. Example 1

Solution : 1. Determine pump suction head.

$$Hs = \frac{(P_1 - P_{f1})(2.31)}{\gamma} + H_1$$

$$\frac{(15 - 2)(2.31)}{0.8} + 15$$

$$= 52 \text{ ft (if kinetic)}$$

or

$$52 \left(\frac{0.8}{2.31}\right)$$

$$= 18 \text{ psig (if positive displacement)}$$

2. Determine pump discharge head.

$$H_D \quad \frac{(P_2 + P_{f2} + P_c)(2.31)}{\gamma} + H_2$$

$$= \frac{(500 + 4 + 0)(2.31)}{0.8} + 80$$

$$= 1564 \text{ ft (if kinetic)}$$

or

$$= 1564 \left(\frac{0.8}{2.31}\right)$$

$$= 542 \text{ psig (if positive displacement)}$$

3. Determine the total dynamic head.

$$TDH = H_D - H_S$$

$$= 1564 \text{ ft} - 52 \text{ ft}$$

$$= 1512 \text{ ft (if kinetic)}$$

or

$$1512 \left(\frac{0.8}{2.31}\right)$$

$$= 524 \text{ psig (if positive displacement)}$$

4. Determine the pump horsepower required.

$$BHp = \frac{(GPM)\,(TDH)\,(\gamma)}{(3960)\,(e)}$$

$$= \frac{(150)\,(1512)\,(0.8)}{(3960)\,(0.52)}$$

$$= 88 \text{ (if kinetic)}$$

or

$$BHp = \frac{(GPM)\,(\Delta P)}{1714\,(e)}$$

$$= \frac{(150)\,(524)\;.}{(1714)\,(0.82)}$$

$$= 56 \text{ (if positive displacement)}$$

5. Determine the electrical energy consumption.

KWHR/ Day = 17.9 (BHp)/ Motor Efficiency

$$= \frac{(17.9)\,(88)}{0.92}$$

$$= 1712 \text{ (if kinetic)}$$

$$\text{KWHR/ Day} = \frac{(17.9)\,(56)}{0.92}$$

$$= 1090 \text{ (if positive displacement)}$$

4. Pump Performance Curves

A pump performance curve is a vital piece of information that can be essential to operators and maintenance personnel alike. Each combination of pump and impeller has a unique set of performance curves. Any new pump should come with installation, operation, and maintenance data and with pump curves. Files should contain curves on every pump in the plant.

Pump curves can be used to select a pump for a set condition, or they can be used to help determine the effects of changes in impeller diameter, speed, and suction lift on horsepower requirements, flow, and efficiency.

There are three basic types of curves used for centrifugal pumps.

- The head capacity curve.
- The efficiency curve.
- The horsepower demand curve.

Some pump curves also include a curve for NPSH. Before looking at specific curves, you must know:

- The speed of the pump.
- The diameter of the pump's impeller.

Figure 1.9 shows a typical pump curve, Notice that a series of curve depicts various impeller diameters. A series of curves also shows efficiency and another brake horsepower. At first glance, this information can appear very confusing. If you analyze each type of curve individually, however, pump performance curves are much easier to understand.

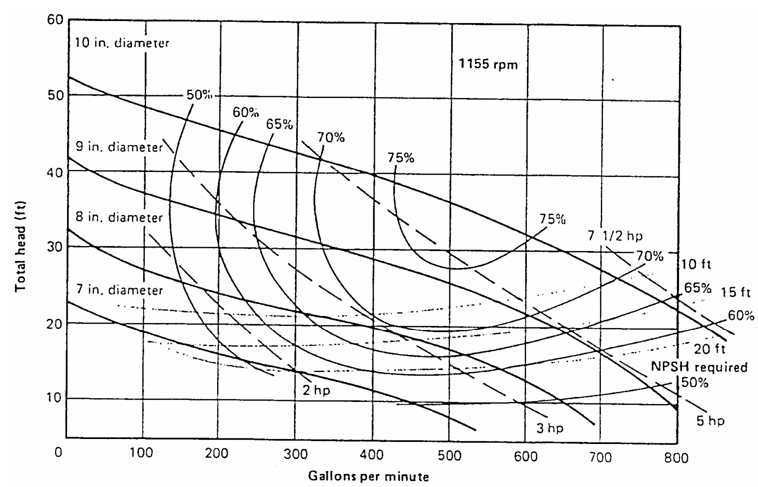


FIGURE 1.9 TYPICAL PUMP CURVE

a. Head Capacity Curves

The head capacity curve is the most basic and useful of all the pump curves. It is a graphic display of the relationship between total head and flow conditions. Notice that the curve shown in Figure 1.10 is for one 9-in impeller and running at one speed-3200 rpm. Notice that for any given head, one and only one flow condition exists, and vice versa. Also, notice that for a given head, a flow can be found and for a given flow, a head can be found.

Refer to Figure 1.10 for the following example. If a head of 300 ft is required, what flow can the pump produce? First, enter the curve form the left at **300 ft.** Continue to the right in a straight line until you meet the 9-in diameter impeller curve. Now move down in a straight line to read a flow 230 gpm.

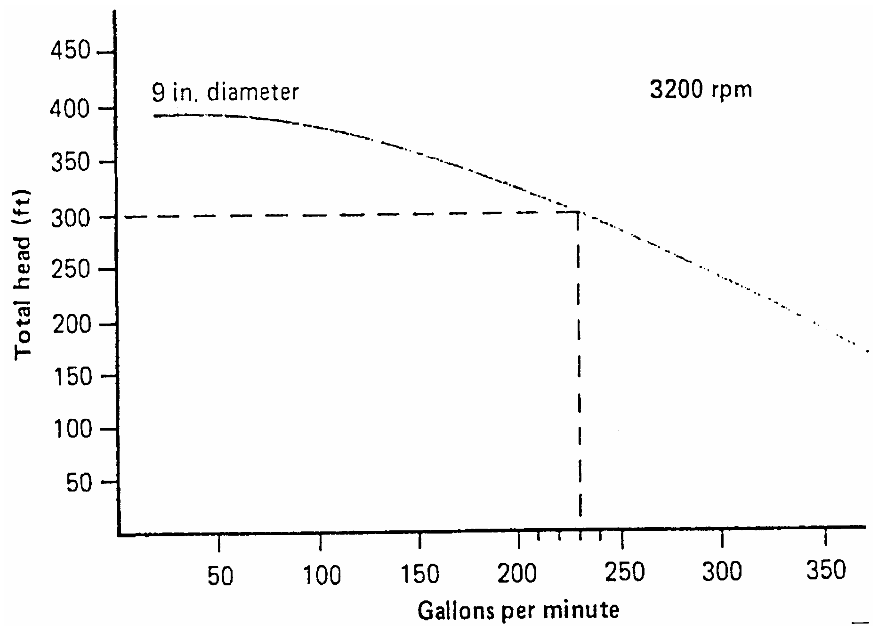


FIGURE 1.10 HEAD CAPACITY CURVE

From the curve, you can see that the maximum total head to be expected is about 380 ft using the 9-in, impeller at 3200 rpm. Remember that this head includes suction lift (if you have one), discharge static pressure, and function losses on both the suction and discharge side of the pump, This maximum pressure or head is reached when the pump is shut down and is referred to as shutdown head.

From the curve, it is easy to see how the pump responds to chances in head. If the head increases, the flow automatically decreases. If the head decreases, the flow increases. The pressure developed by a pump is also dependent on its speed. For that reason, as speed decreases, the head capacity curve sinks straight down toward the bottom of the graph. As speed increases, the curves rise toward the top of the graph, The curve always maintains its same basic shape. The same is true for the impeller diameter. As diameter decreases, the curve sinks.

b. <u>Efficiency Curves (Figure 1.11)</u>

It is important that pumps be operated so that the most amount of work can be done for a given amount of expended energy. The efficiency curve will tell you at what total head the best energy transfer will take place.

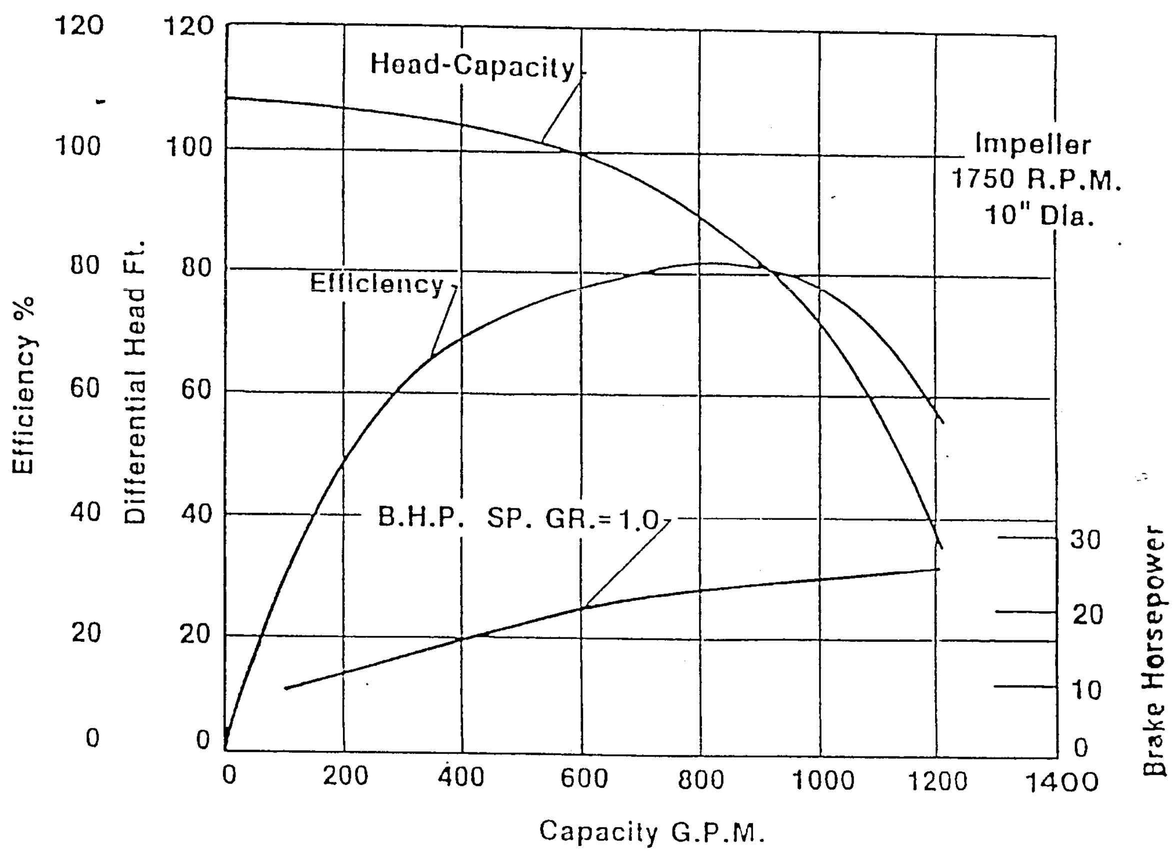


FIGURE 1.11 CHARACTERISTICS CURVES AT CONSTANT SPEED

c. Horsepower Curves (Figure 1.11)

The horsepower curve gives information on the horsepower required by the pump, not necessarily the horsepower output of the motor. For example, suppose you require 17 hp in a certain pumping situation. Motors are manufactured in 15 hp and 20 hp, but not 17 hp. Therefore, a 20 hp motor is needed while pumping however, the motor would be required to produce only the 17 hp. The horsepower requirement increases as flow increases. Maximum horsepower is required at maximum discharge. Minimum horsepower is required when the discharge is closed.

5. **Pump Classification**

A. Kinetic Pumps

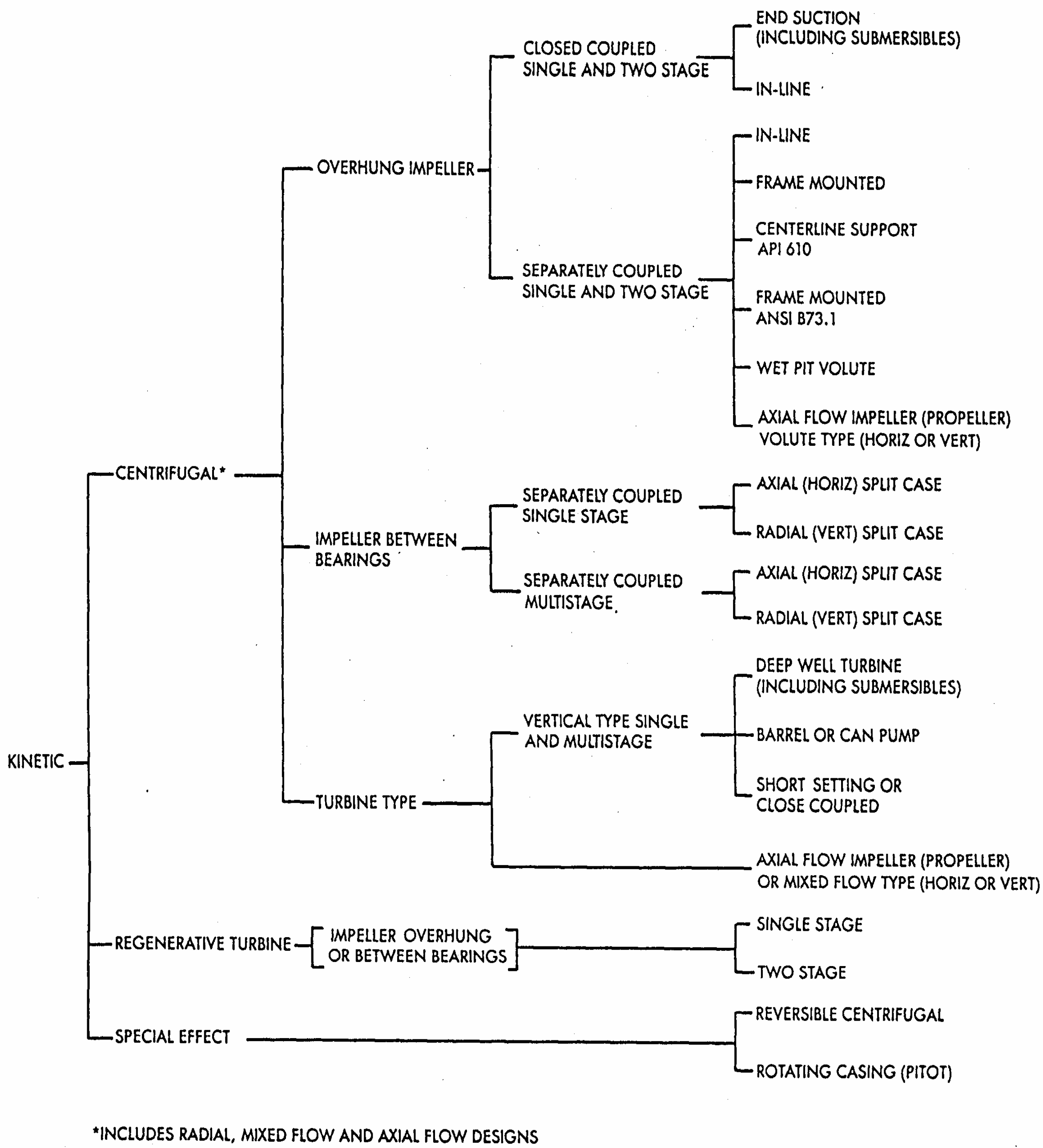


Figure 1. Kinetic Energy Pump Classifications (courtesy Hydraulic Institute)

B. Positive Displacement Pumps

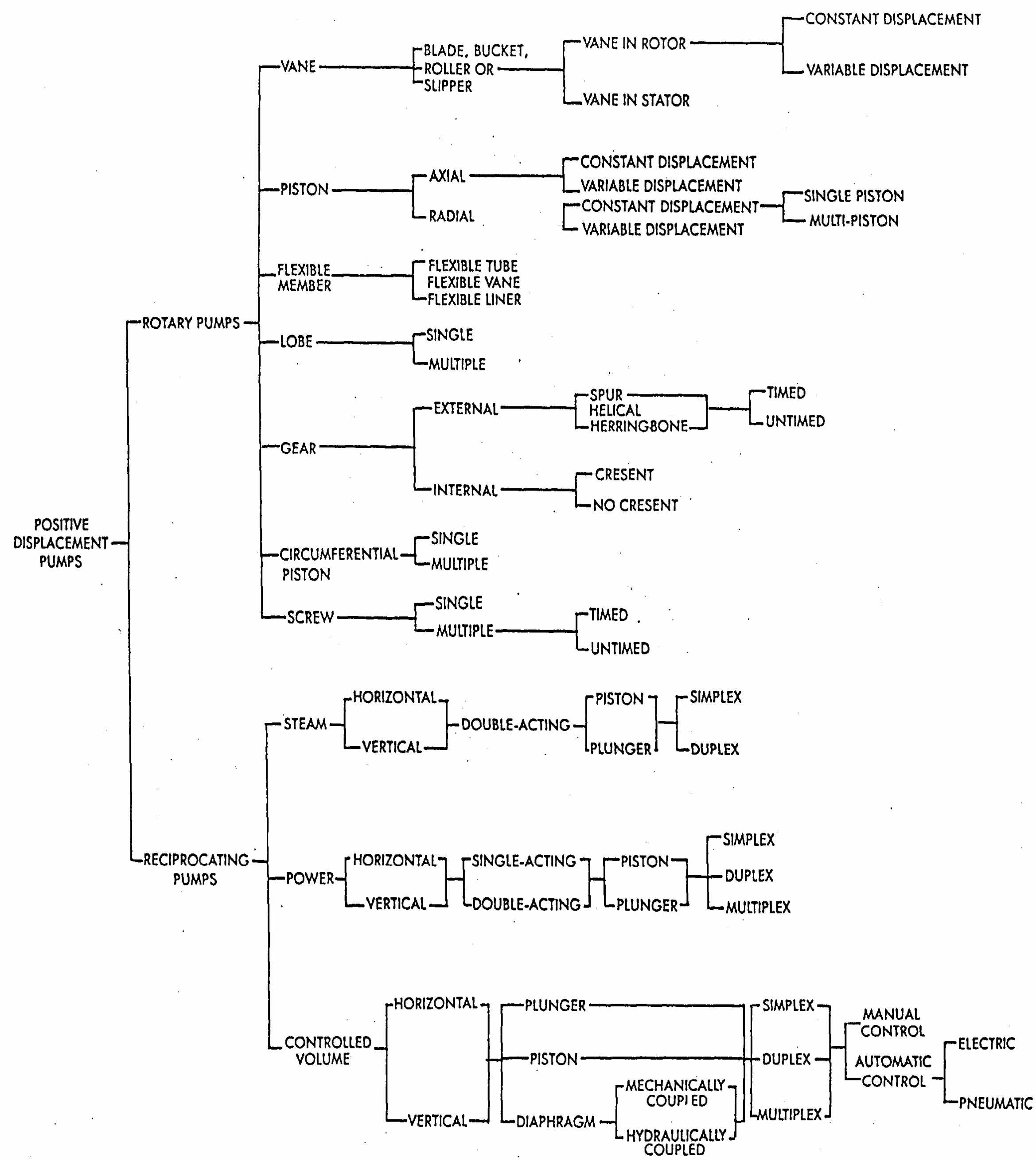
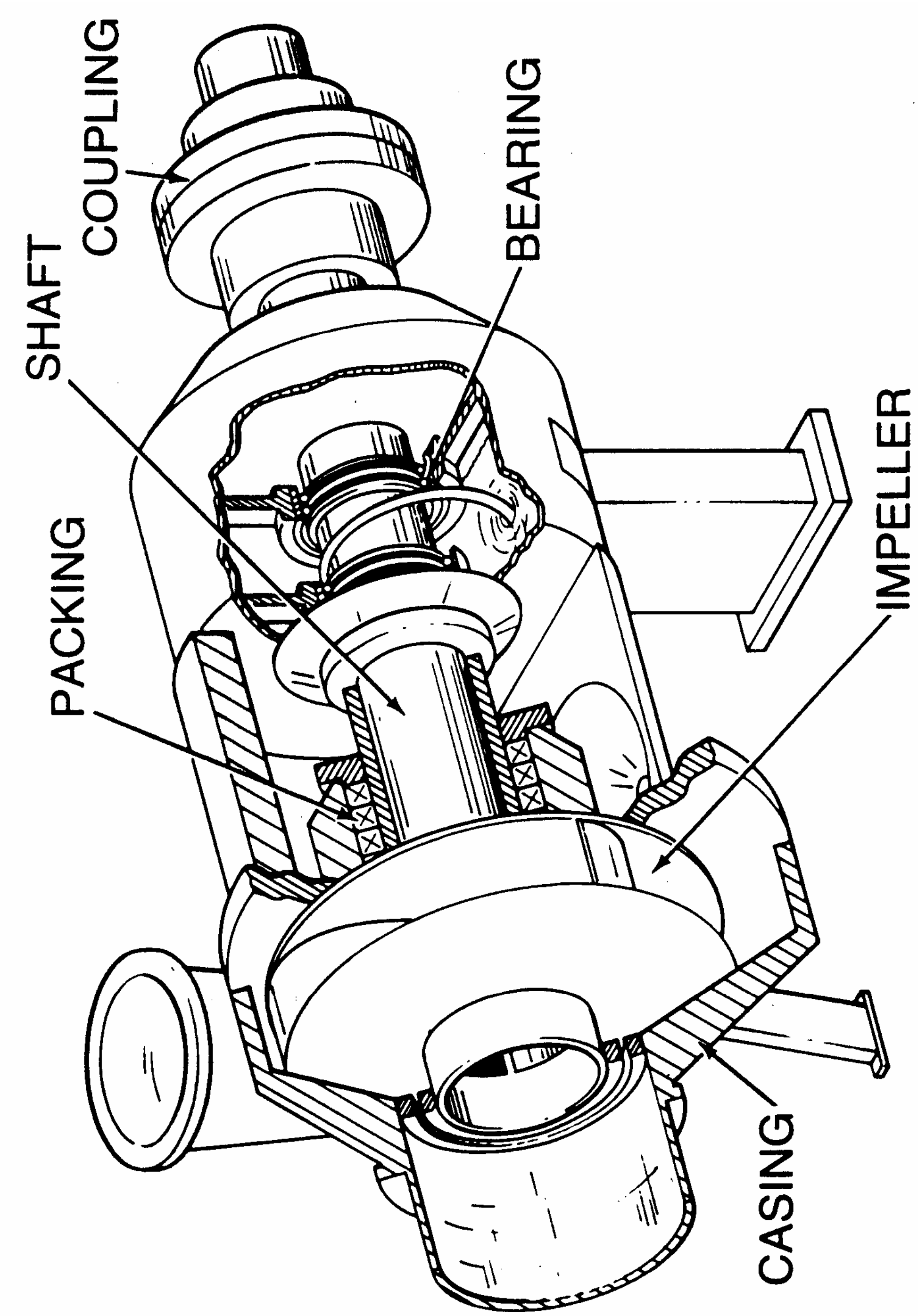
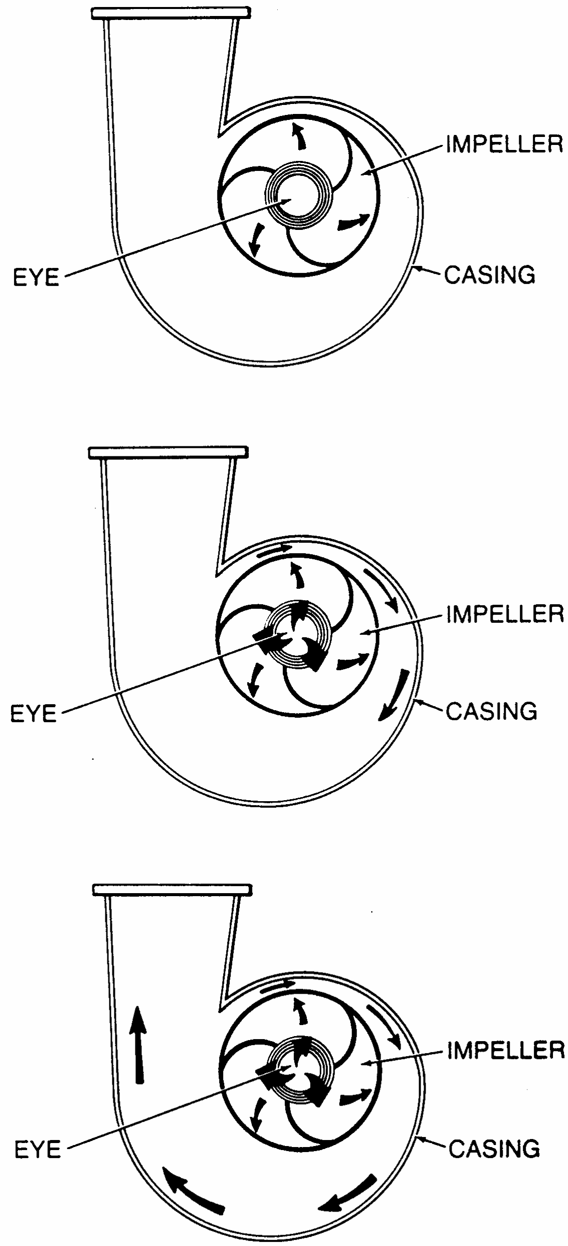


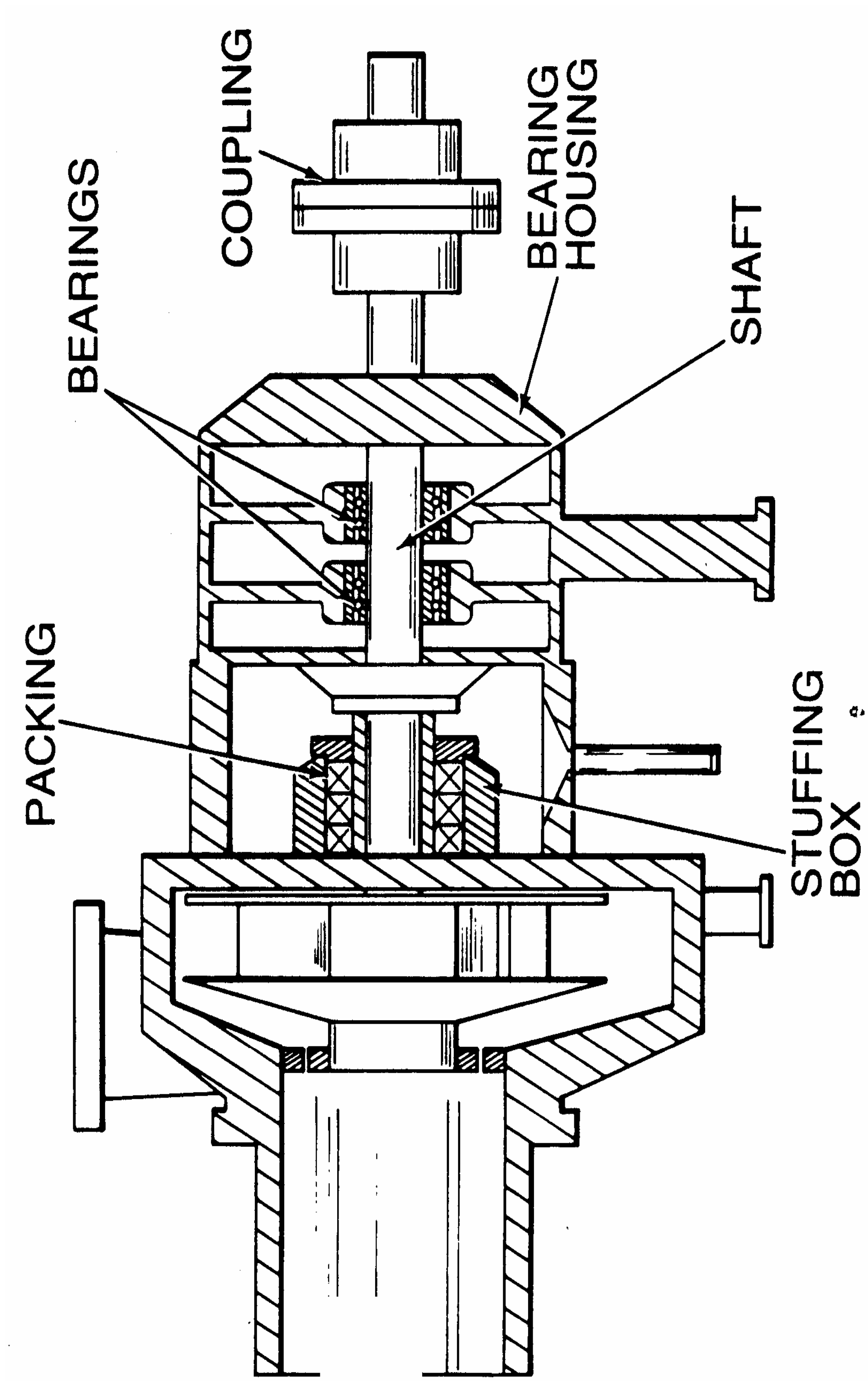
Figure 2. Positive Displacement Pump Classifications
(courtesy Hydraulic Institute)



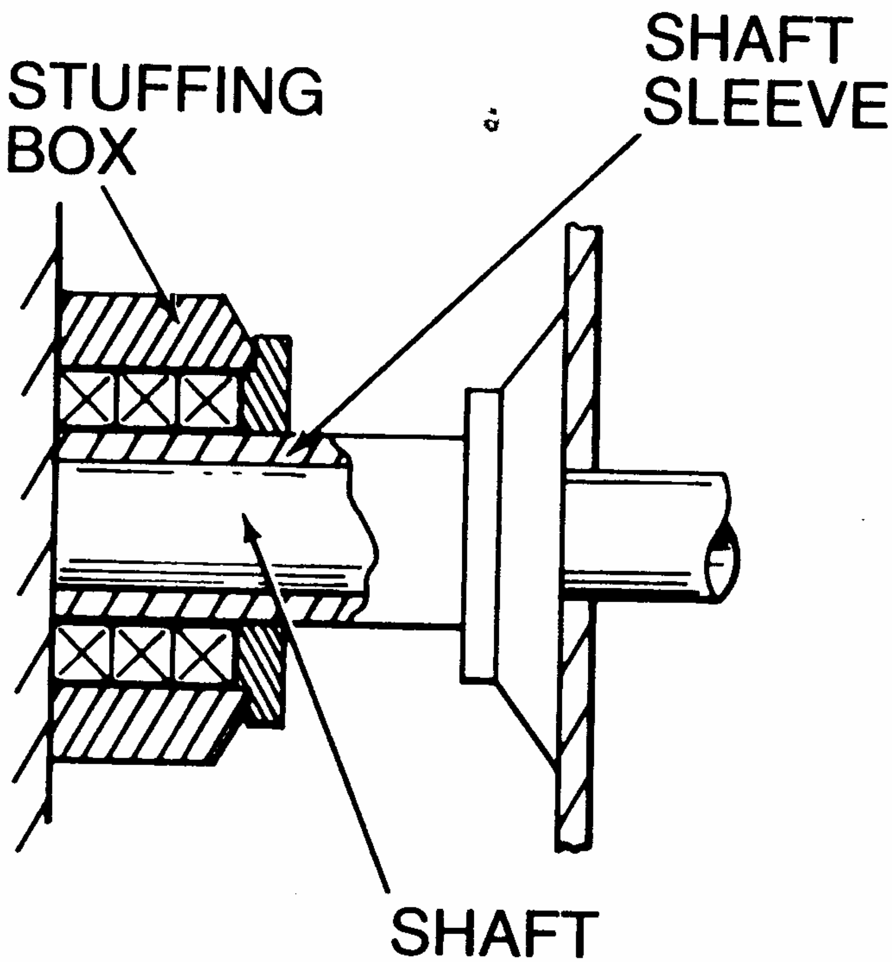
TYPICAL CENTRIFUGAL PUMP

OPERATING PRINCIPLE

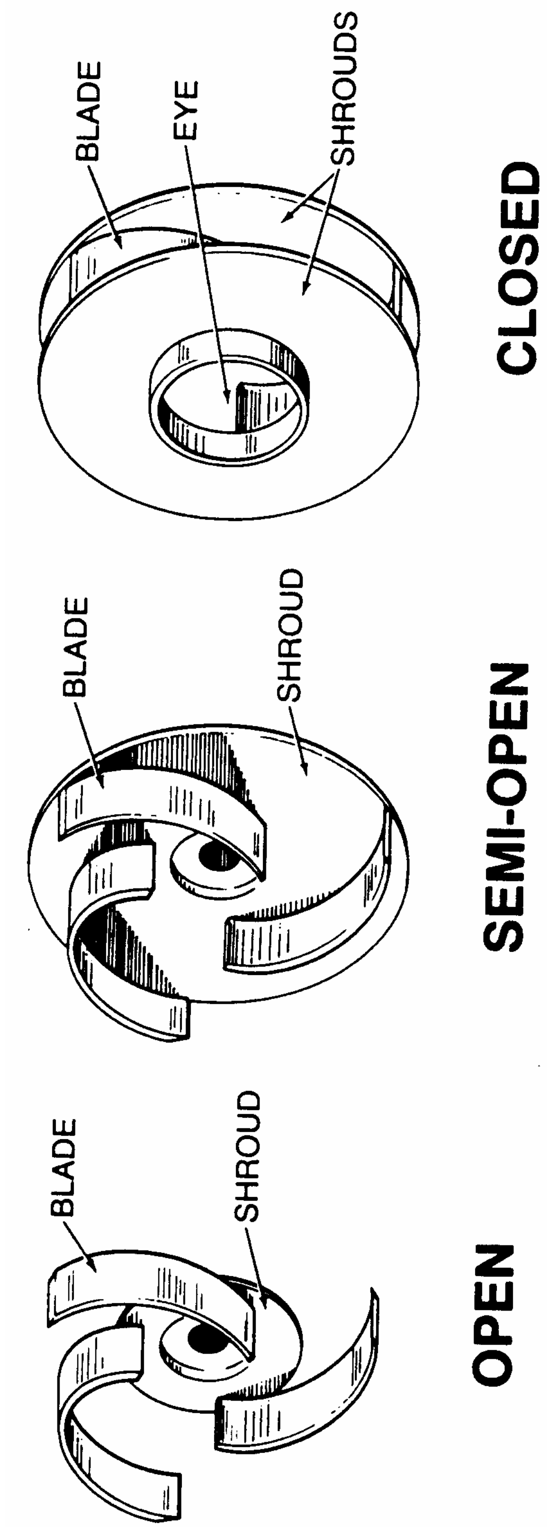


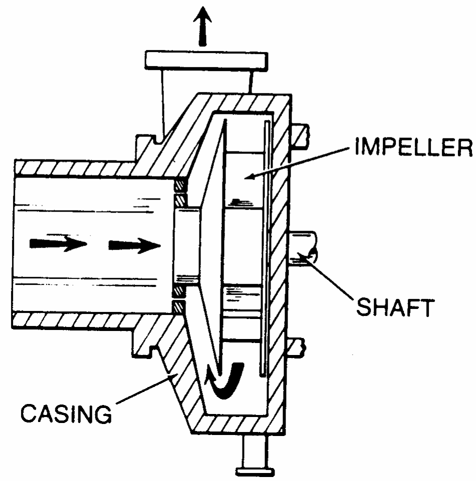


PUMP SIDE VIEW

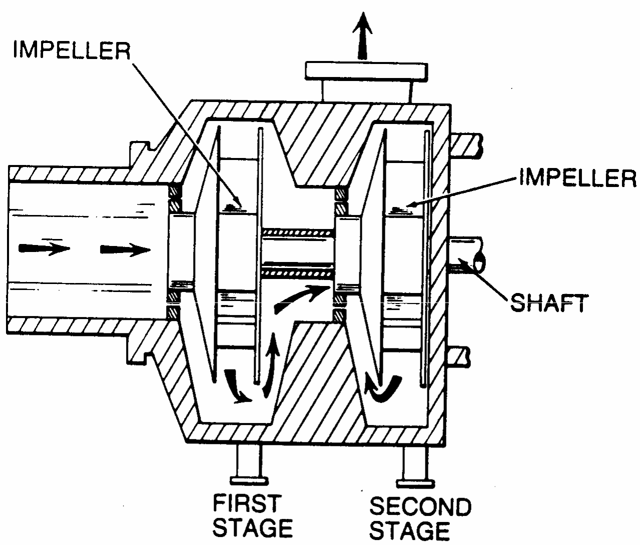


PUMP SHAFT WITH SLEEVE

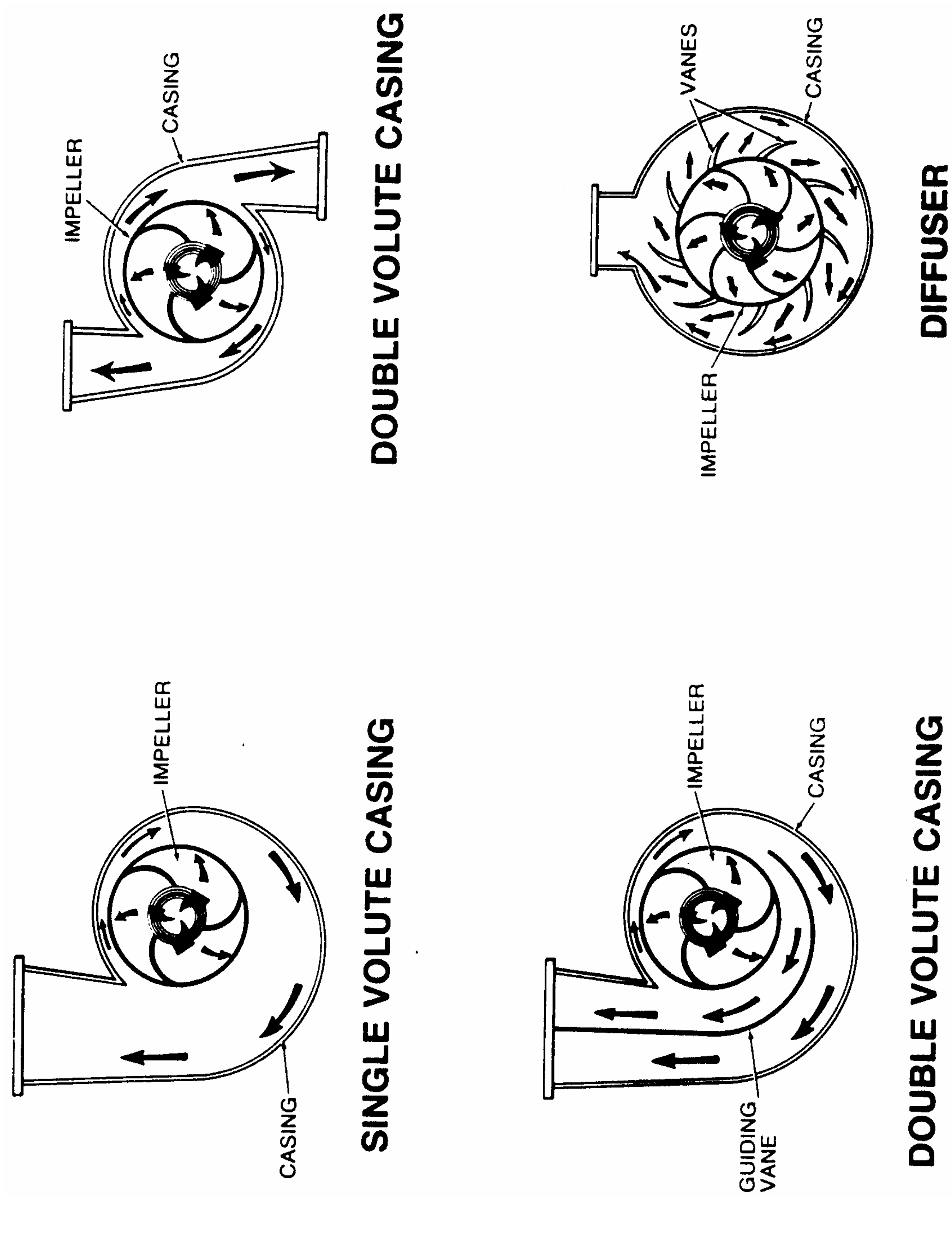

BLADE
EYE
SHROUDS
CLOSED
BLADE
SHROUD
SEMI-OPEN
BLADE
SHROUD
OPEN

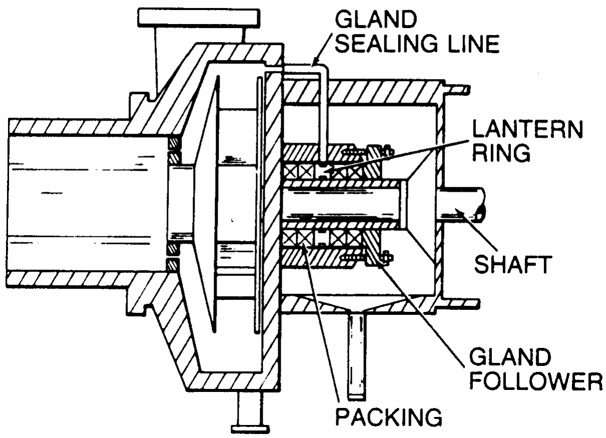


SINGLE-STAGE PUMP

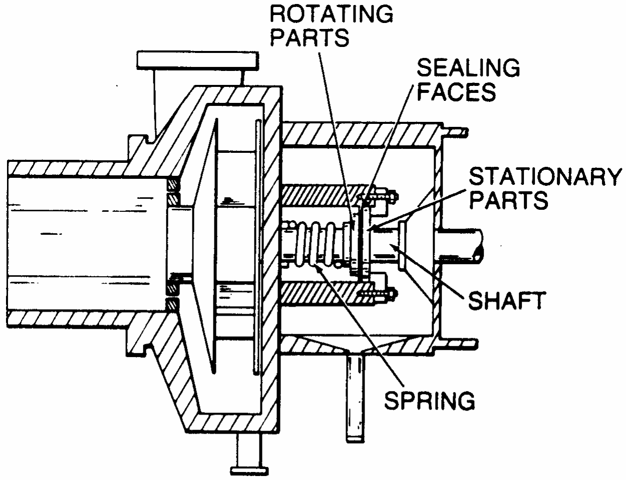


MULTI-STAGE PUMP

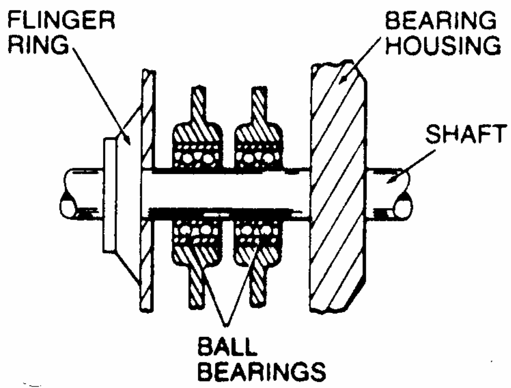

IMPELLER
CASING
SINGLE VOLUTE CASING
IMPELLER
CASING
GUIDING VANE
DOUBLE VOLUTE CASING
IMPELLER
CASING
DOUBLE VOLUTE CASING
VANES
CASING
IMPELLER
DIFFUSER



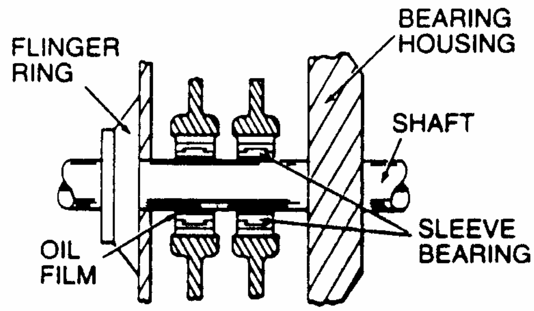
STUFFING BOX



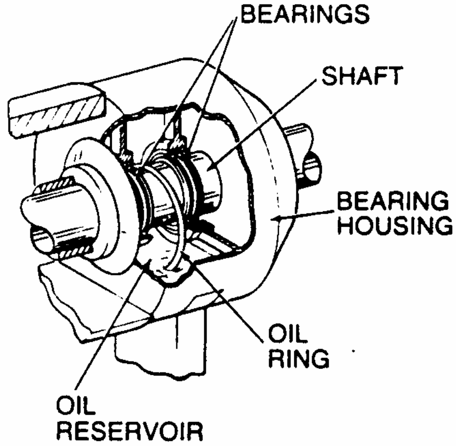
MECHANICAL SEAL



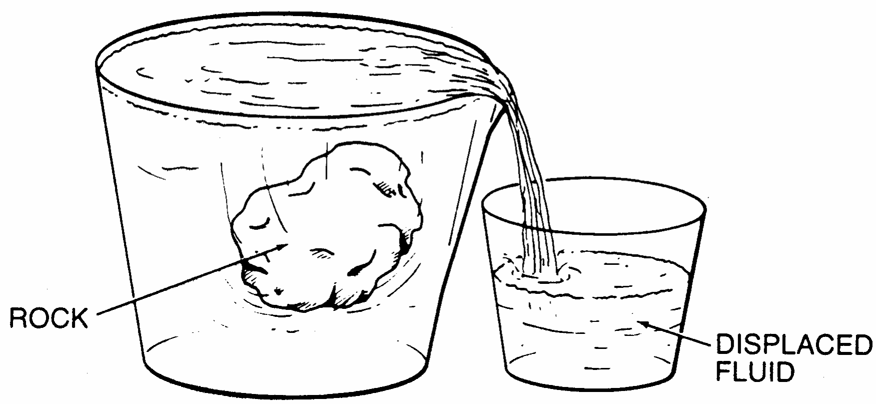
BALL BEARINGS



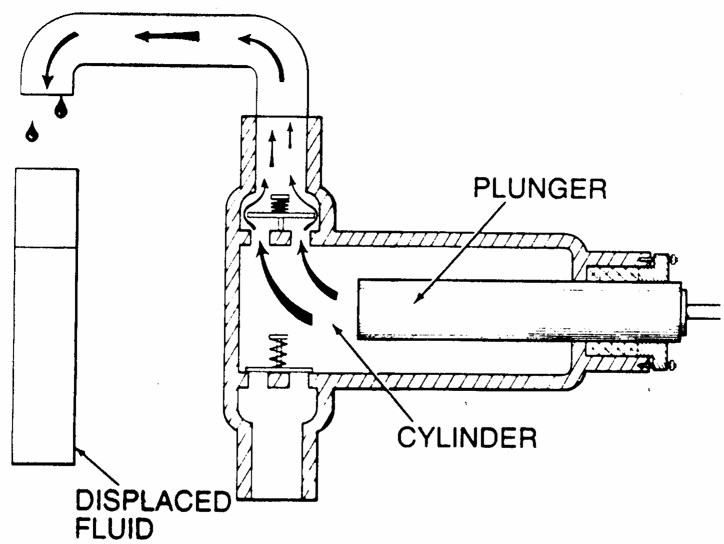
SLEEVE BEARINGS



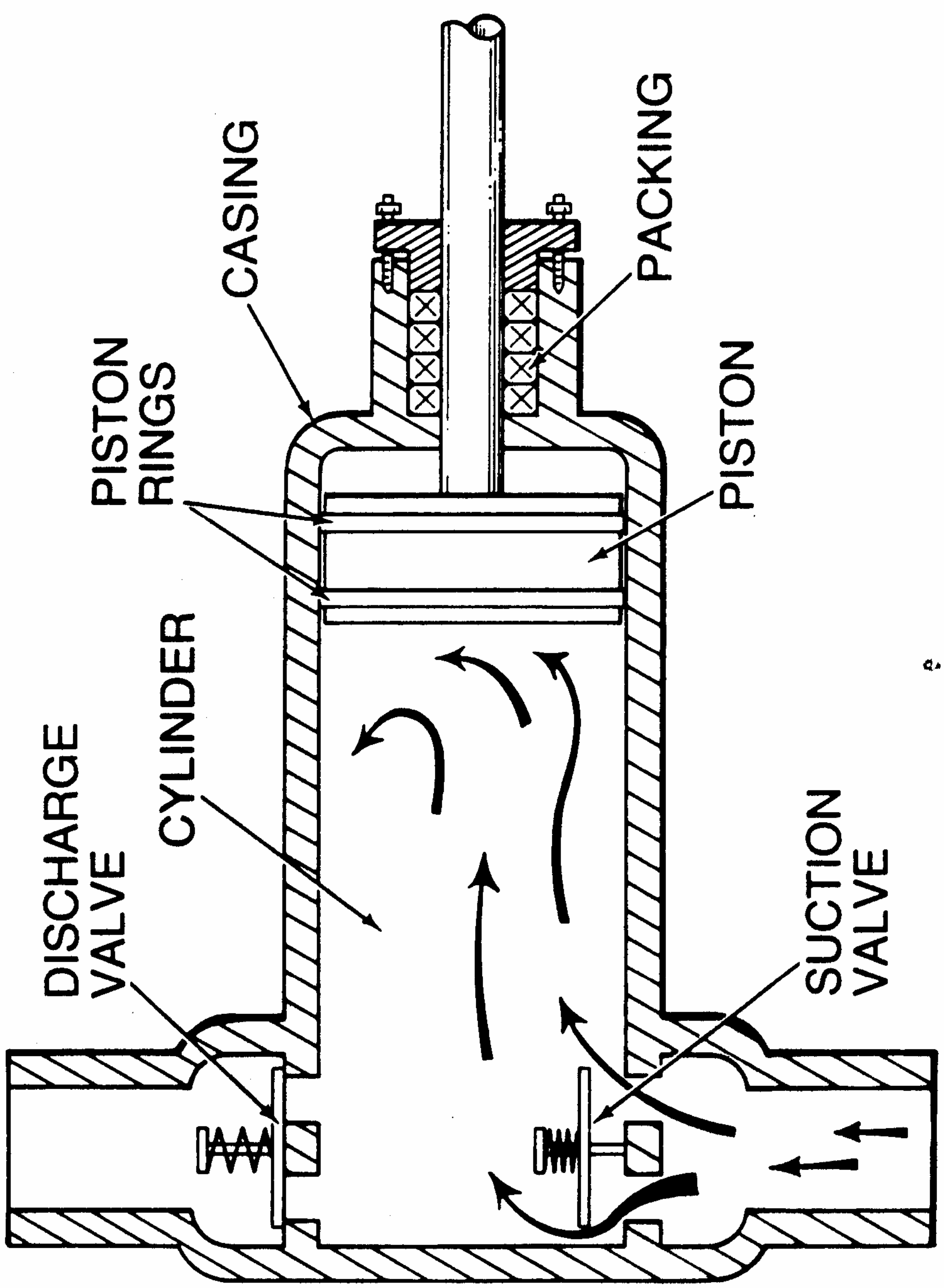
BEARING LUBRICATION



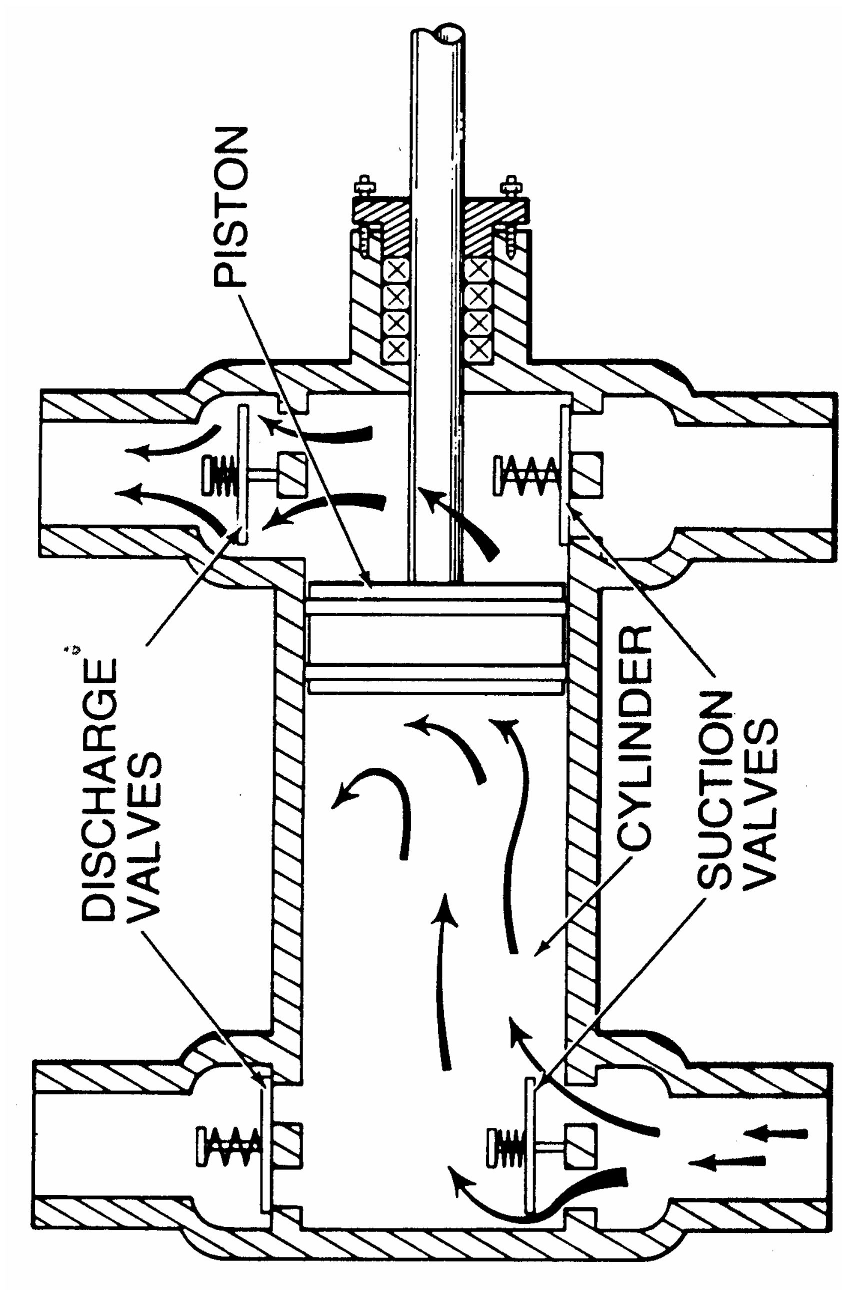
POSITIVE DISPLACEMENT



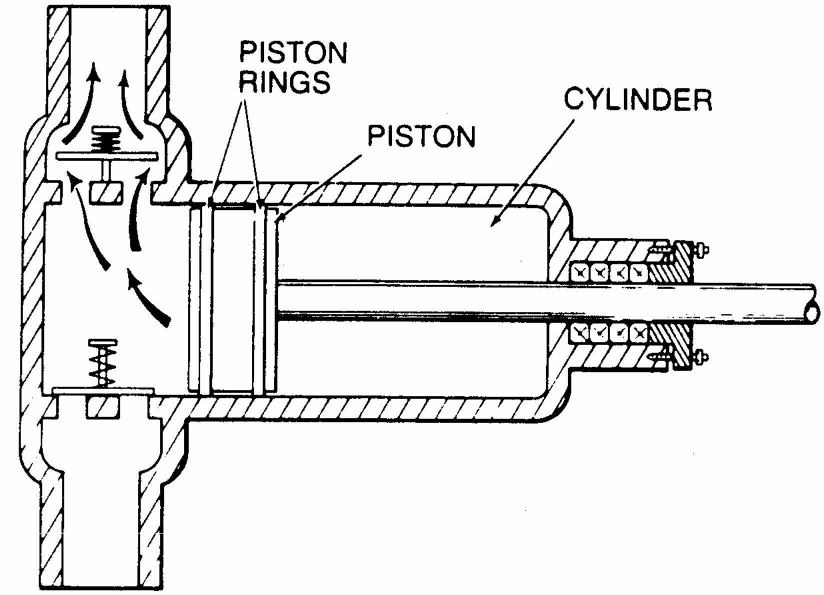
POSITIVE DISPLACEMENT PUMP


CASING
PACKING
PISTON RINGS
PISTON
CYLINDER
DISCHARGE VALVE
SUCTION VALVE

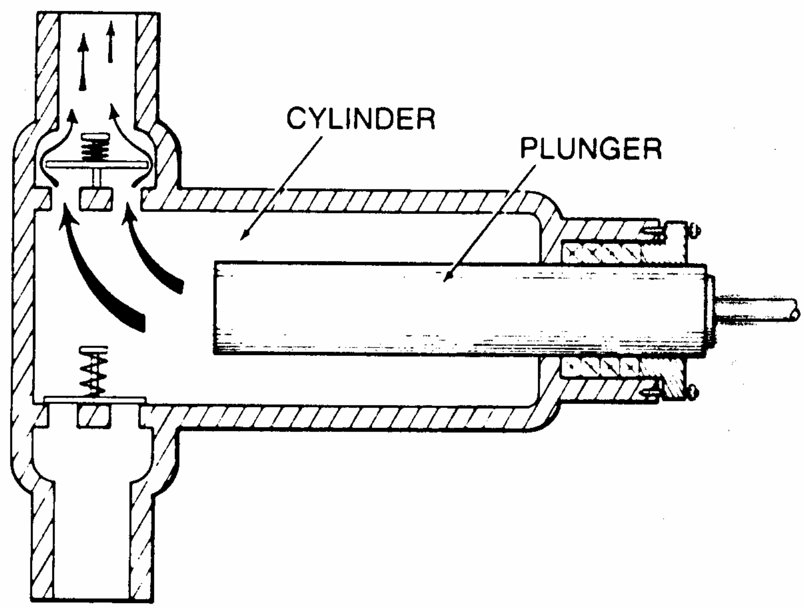
PISTON PUMP



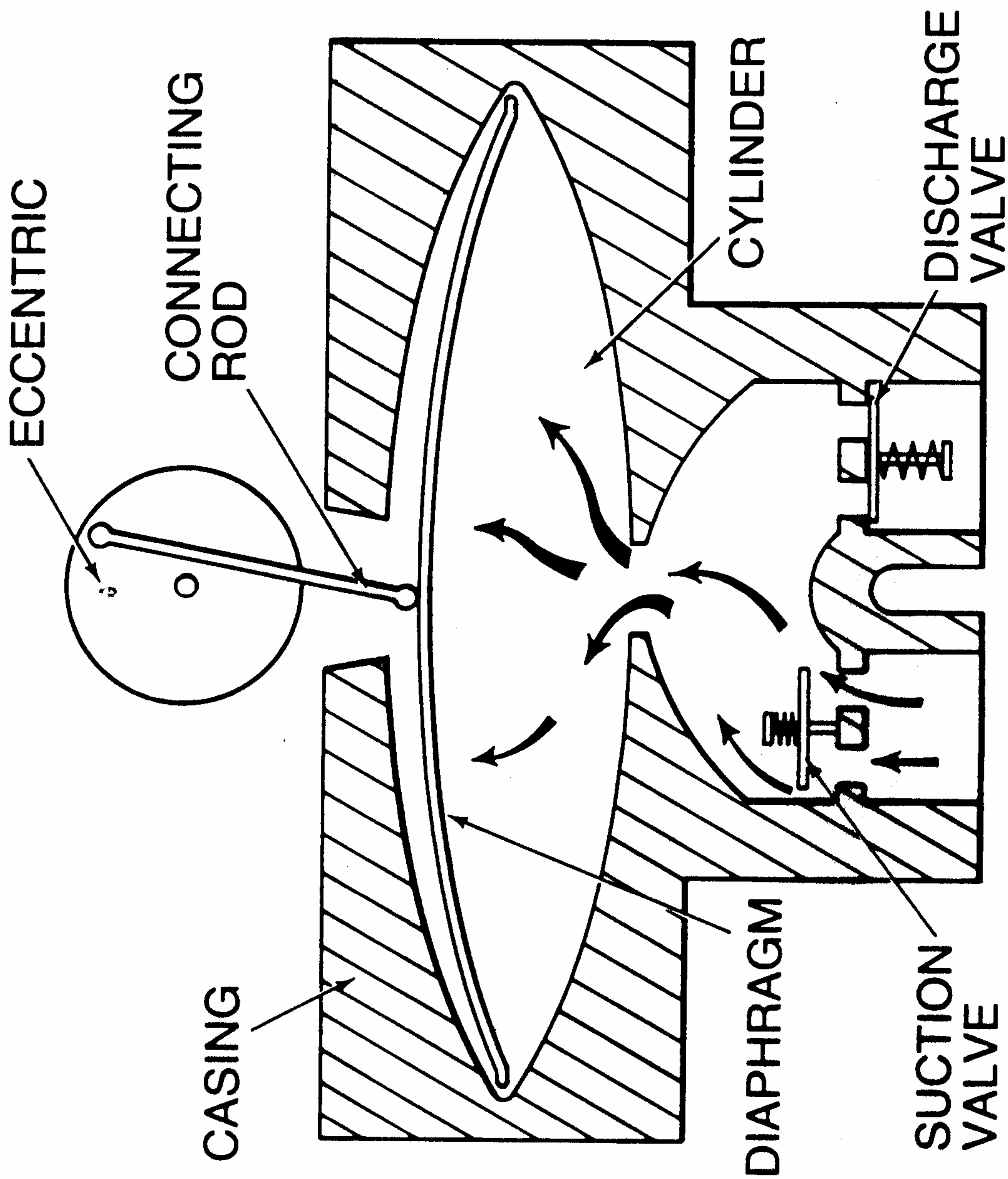
DOUBLE-ACTING PUMP



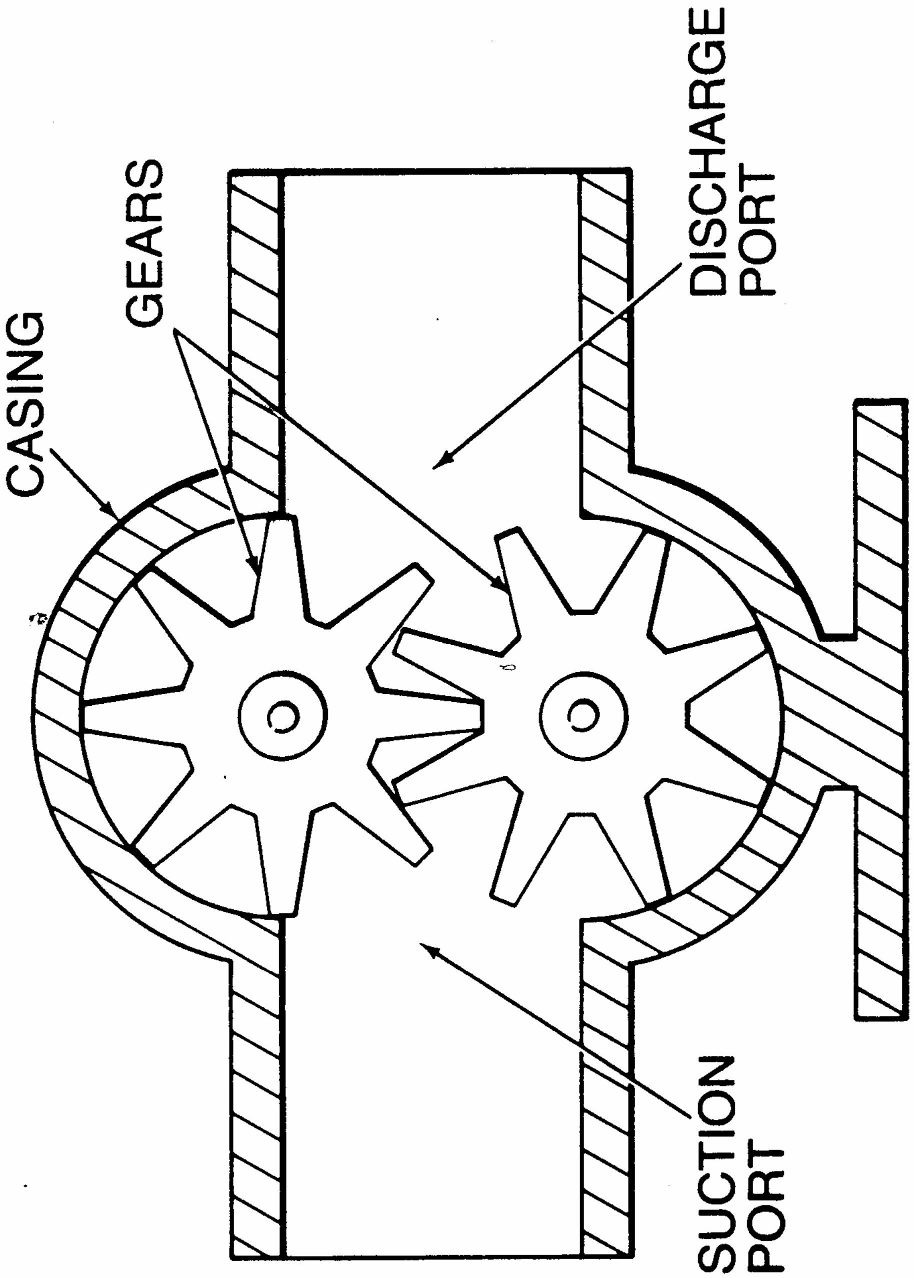
PISTON PUMP



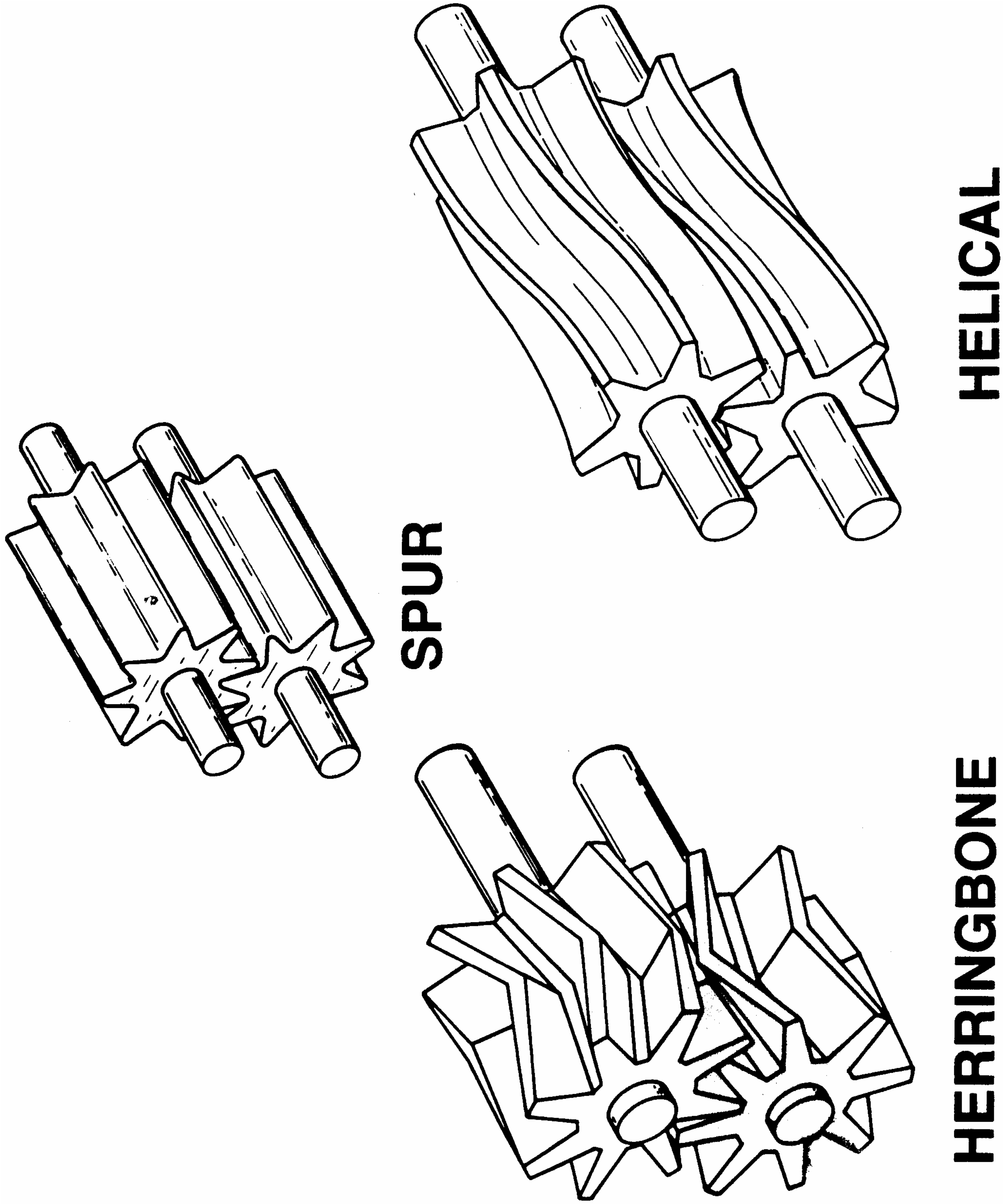
PLUNGER PUMP

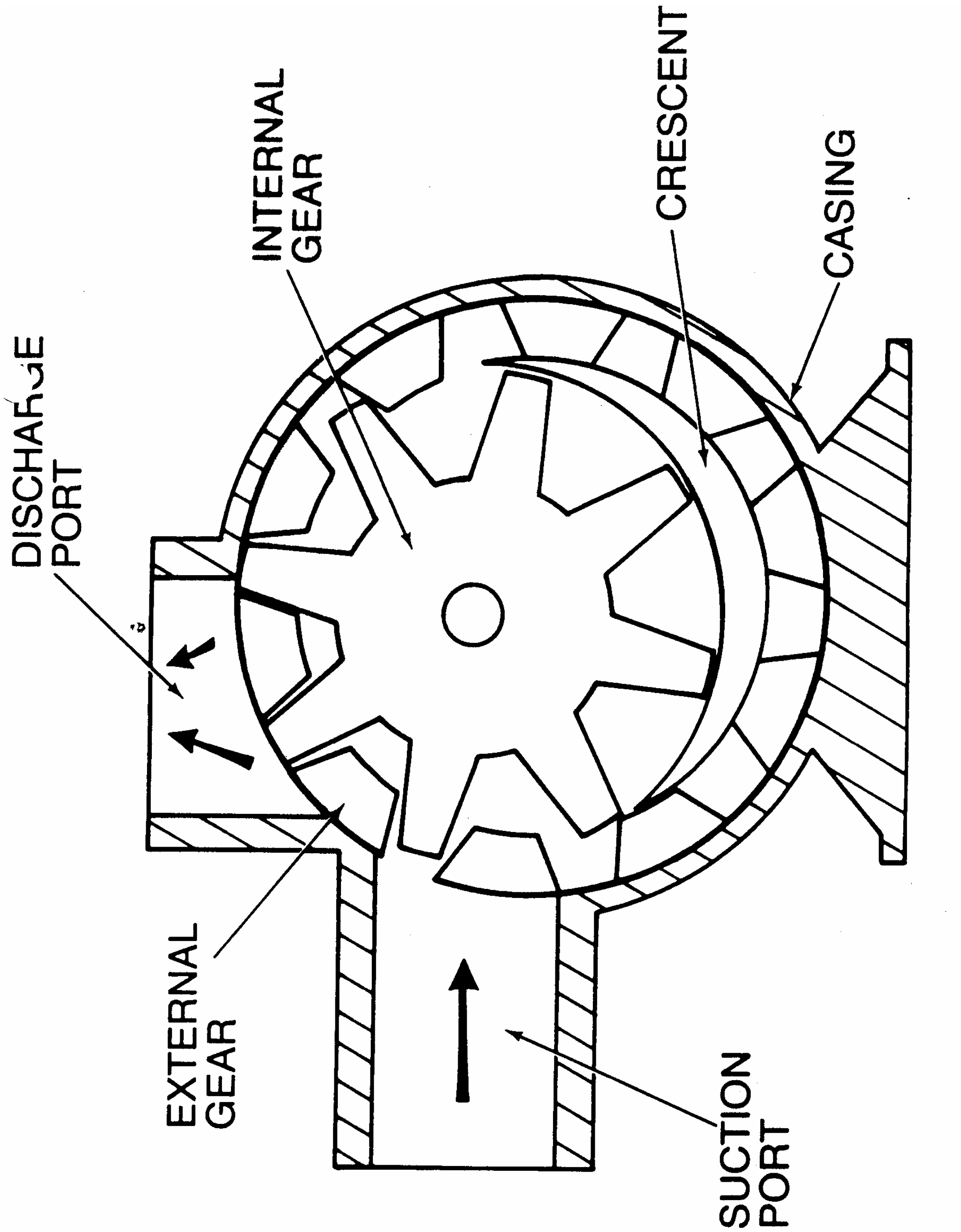


DIAPHRAGM PUMP

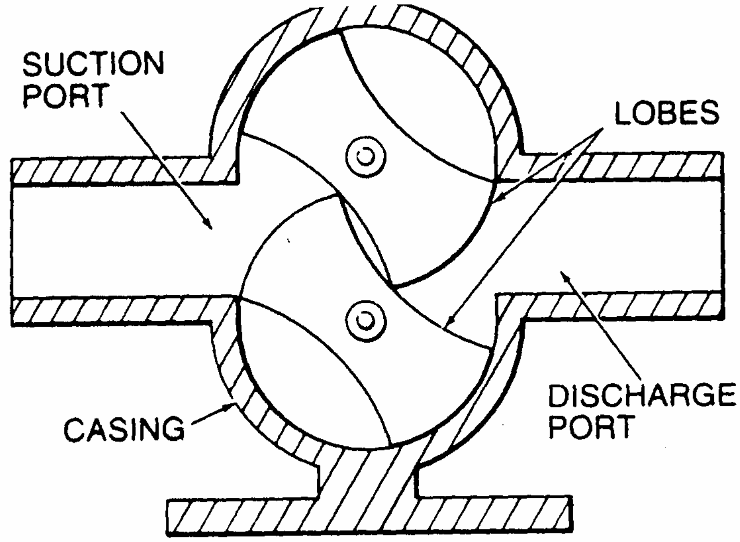


EXTERNAL GEAR PUMP

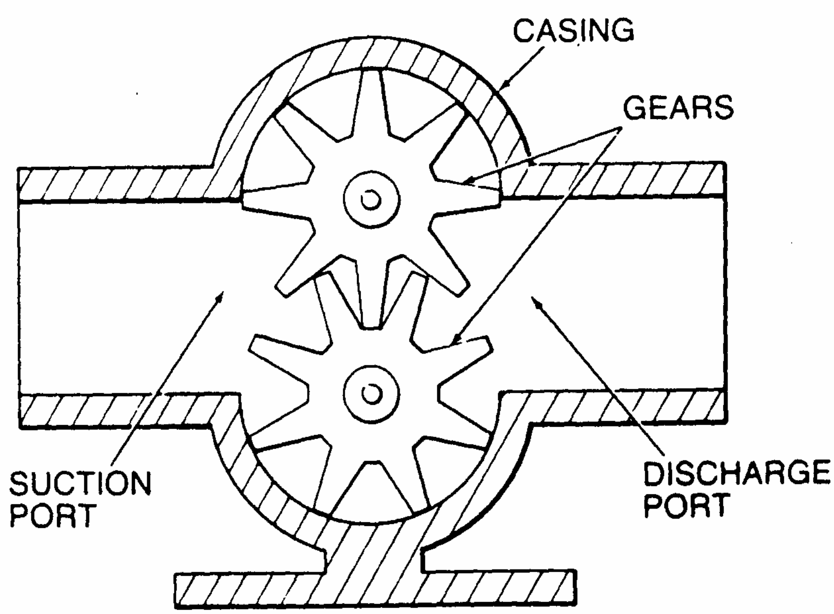

HELICAL
SPUR
HERRINGBONE



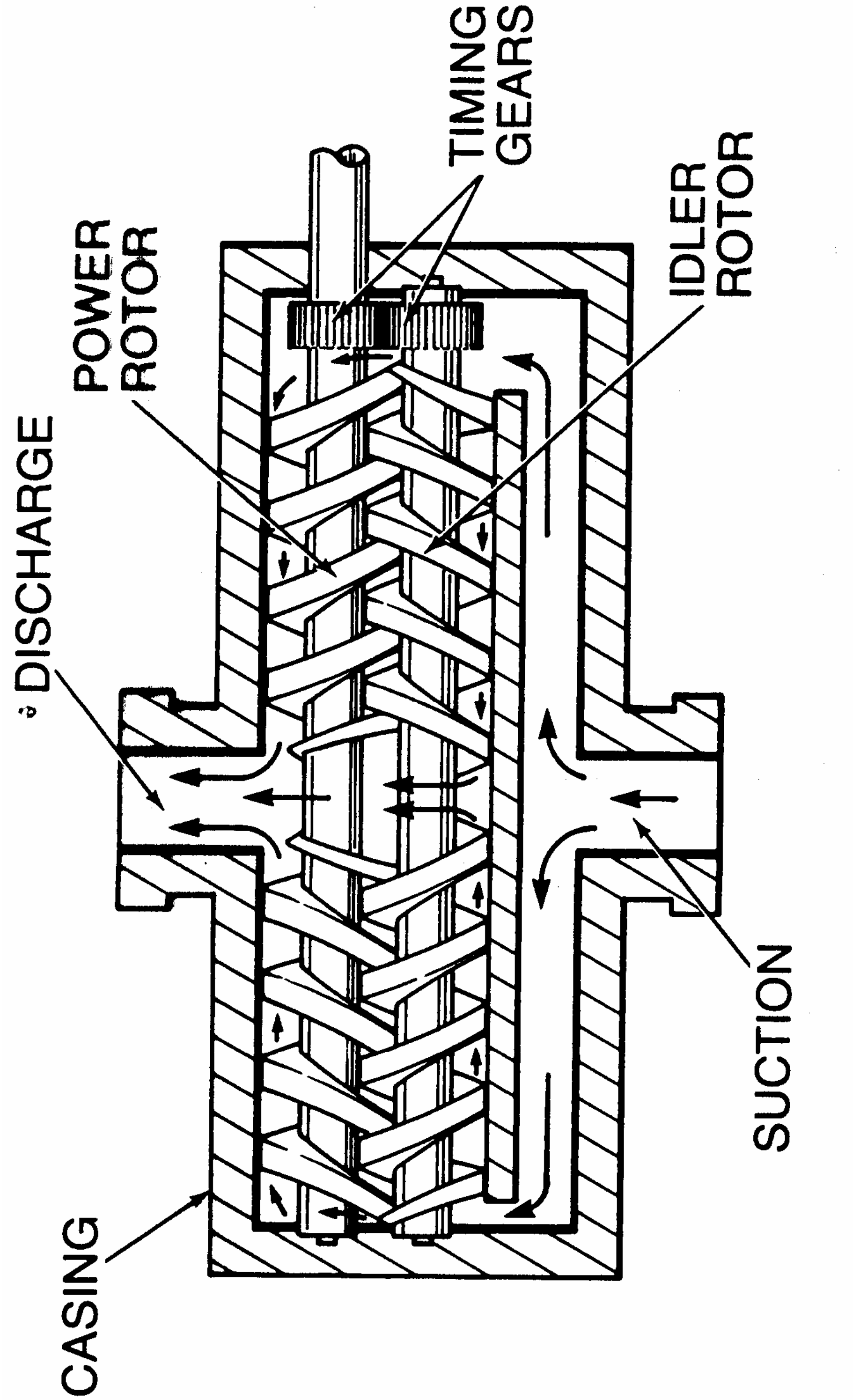
INTERNAL GEAR PUMP



LOBE PUMP



EXTERNAL GEAR PUMP


TIMING GEARS
POWER ROTOR
IDLER ROTOR
DISCHARGE
CASING
SUCTION

SCREW PUMP

6. Troubleshooting - Centrifugal Pumps

The following is a guide for troubleshooting centrifugal pumps and pump systems.

1. Failure to Deliver Liquid

   A. Pump not primed.

   B. Insufficient speed.

   C. Discharge head too high.

   D. Suction lift too high (over 15 feet), check with vacuum gauge.

   E. Impeller passages partially clogged.

   F. Wrong direction of rotation.

2. Pump Overloads Driver

   A. Total head LOWER than rating -pumping too much liquid.

   B. Liquid pumped of different specific gravity and viscosity than that for which pump is rated.

   C. Mechanical defects:

   1. Shaft bent.
   2. Rotating element binds.
   3. Worn bearings.

3. Insufficient Pressure

   A. Speed too low.

   B. Air in liquid.

   C. Mechanical defects.

   * Wearing rings worn.
   * Impeller damaged.
   * Internal leakage due to defective gasket.

4. <u>Insufficient Capacity</u>

A. Air leaks in suction or stuffing boxes.

B. Speed too low.

C. Total head higher than that for which pump is rated.

D. Suction lift too high (over 15 feet), check with vacuum gauge.

E. Impeller passages partially clogged.

F. Insufficient suction head for hot liquid.

G. Mechanical defects.

- Wearing rings worn.
- Impeller damaged.
- Internal leakage due to defective gasket.

H. Foot valve too small or restricted by trash.

1. Foot valve or suction pipe not immersed deeply enough.

J. Suction strainer blocked.

5. <u>Pump Loses liquid after Starting</u>

A. Leaky suction line.

B. Suction lift too high (over 15 feet).

C. Air or gases in liquid.

D. Suction strainer plugged.

6. <u>Pump Vibrates</u>

A. Misalignment.

B. Foundation not sufficiently rigid (grounding broken).

C. Impeller partially clogged, causing, imbalance.

D. Mechanical defects.

* Shaft bent.
* Rotating element binds.
* Worn bearings.

E. Vaporizing in suction

F. Excessive capacity.

G. Suction strainer plugged.

7. Surges in Performance

A. Air leak in suction line.

B. Air pocket in suction line.

C. Not enough NPSH available.

D. Air or gases in liquid.

E. Impeller plugged.

8. Excessive Power Consumption

A. Speed variation - cheek whether or not motor is across the line and receives full voltage.

B. Head too low - pumping too much liquid.

C. Specific gravity or viscosity of liquid pumped is too high.

D. Mechanical defects:

* Shaft bent.
* Rotating element binding.
* Wrong rotation.

9. Noisy Pump Operation

A. Hydraulic noise:

* Cavitation.
* Insufficient NPSH.
* Suction lift too high.
* Air in liquid.

B. Mechanical defects:

* Shaft bent.
* Bearing worn.
* Rotating parts binding.

7. Routine Checks and Troubleshooting - Positive Displacement Pumps

I) Routine Checks on displacement pumps are:

- Check pump suction & discharge pressure.
- Check pump leakage.
- Check lube oil system.
- Check cooling system.
- Check safeguarding system.
- Check suction strainer PD

I.1) Start-up and Stop

A. Prestart Up and Operating Cheeks

1. The set stroking length of the pump must be within its operating range, therefore on no account must the manual handwheel be turned passed its maximum position or its zero mark.

2. Check that any line valves are in their correct operating position before start up.

3. Check that the pump gearbox oil is at the correct level before start up.

4. Turn the perspex cover over on the diaphragm pump head hydraulic fluid oil reservoir, so that the vent opens up into the reservoir. Check that the level of hydraulic fluid covers the vent and replenishing valve in the reservoir.

5. Check that the drive motor rotates in the direction of the arrow on the motor cowling. This can be done by flicking the motor starter switch and watching the direction of the motor fan. If the direction of rotation is incorrect, switch two wires to give reversed direction of rotation.

6. If suction or discharge dampers are fitted to the process line, check that the pre-charge is correct for operation before start up.

7. If a line pressure gauge is fitted to the pump discharge line, check the reading is in accordance with that shown on the applicable pump data sheet. If the discharge pressure is excessive or rises after start up, check for blocked pipework or shut-off line valve. It is imperative that the fault is rectified immediately to ensure that the discharge pressure does not exceed the safe working pressure of the pump head.

B. Start up Procedure

Once the pumps have been checked for their general integrity, and all nuts and bolts have been checked for tightness, the pumps are ready for start up.

1. Set the stroke length of the metering pumps to zero. This will ensure that no fluid is pumped upon start up.

2. Start the pump drive motor running. Run for a period, thirty minutes will suffice, check for any excessive temperature rise.

3. Stop the drive motor. Recheck the oil level.

4. Increase the stroke length to half of the maximum output capacity, start the drive motor.

5. Once a fluid flow has been established in the discharge line set the stroke length to ensure quick venting of the process line.

6. If no fluid is being pumped, or if output is low refer to the general fault finding chart.

7. Check all connections, flanges, valves, etc. are leak free and tight.

C. Routine Shut Down

1. No attention is required to the pumps if it is for short term shut down. The pump can be shut down and restarted as a normal process function.

2. If the shut down is for a long period, or maintenance then any line isolation valves should be closed and the motor isolated.

I.2) Operating Problems in Reciprocating Pumps

Operating problems in reciprocating pumps are typically related to packing wear along with the plunger valve wear, and cavitation.

The biggest maintenance problem on most reciprocating pumps is packing. Although the life of standard packing in a power pump is about 2,500 h, some installations with special stuffing-box arrangements have experienced a life of more than 18,000 h, at discharge pressure up to 28 Kpa.

Short packing life can result from any of the following conditions:

1. Improper packing for the application
2. Insufficient lubrication
3. Misalignment of plunger (or rod) with stuffing box.
4. Worn plunger, rod, stuffing box bore or stuffing-box bushings
5. Packing gland too tight or too loose
6. High speed or high pressure.
7. High or low temperature of pumpage.
8. Excessive friction (too much packing in box).
9. Packing running dry (pumping chamber gas-bound).
10. Shock conditions arising from entrained gas or cavitation, broken or faulty valve springs, or system problems.
11. Solids from the pumpage, environment or lubricant.
12. Improper packing installation or back-in (where required).
13. Icing caused by volatile liquids that refrigerate and from ice crystals upon leakage to atmosphere, or by pumping liquids at temperatures below 0°C.

As is evident from these conditions, short packing life can indicate problems elsewhere in the pump or system.

To achieve a low leakage rate, the clearance between the plunger (or rod) and packing must be essentially zero. This requires that the sealing rings be relatively soft and pliant. Because the packing is pliant, it tends to flow into the stuffing-box clearances, especially between the plunger and follower bushing. If this bushing does not provide an effective barrier, the packing will extrude, and leakage will increase.

A pressure gradient occurs across the packing rings. The last ring of packing adjacent to the gland-follower bushing will experience the largest axial loading of all rings resulting in greater deformation, tighter scaling and, therefore, the largest pressure drop. Most packing failures originate at this critical sealing point.

Because this last ring of packing is the most critical, does the most sealing, and generates the most friction, it requires more lubrication than do the others. In the non-lubricated arrangement, this ring must rely on the surface of the plunger to drag some of the pumpage back to it in order to provide cooling and lubrication.

Because the last ring of packing requires more lubrication than do the others, lubrication of the packing from the atmospheric side is often more effective than injection of oil into a lantern ring located in the center of the packing. Care must be exercised to get the lubricant onto the plunger surface and close enough to the last ring, so that the stroke of the plunger will carry the lubricant under the ring.

Soaking the packing in oil prior to installation will enhance a proper break-in and increase packing life.

During the first few hours of pump operation following repacking, each stuffing box should be monitored for temperature. It is normal for some boxes to run warmer than others - as much as 30'C above the pumping temperature. Only if this exceeds the maximum temperature rating of the packing are steps required to reduce box temperature.

The best lubricant for most installations equipped with stuffing-box lubricators has been found to be steam-cylinder oil. This oil is compounded with tallow, which gives it a tenacity for the plunger surface and make it ideal for providing a lubricating wedge between the plunger and packing.

I.3) Trouble Shooting - Reciprocating Pumps

1. Low volumetric Efficiency (Failure to Deliver Rated Capacity and Pressure)

* Air or vapor pocket in suction line.
* Air or vapor trapped in above suction valves.
* Air leak in stuffing box packing
* Air leak in suction piping, cross bolts in pump suction manifold.

* Air or gases entrained in liquid.
* Foreign object holding pump valve open.
* Loose valve covers or cylinder heads.
* Worn valves and seats.
* Safety relief valve not holding pressure or partially open.
* Worn liners, piston rings or plungers.
* Bypass valve open or not holding pressure.
* Blown liner gasket.
* NPSH available not sufficient.
* Fluid bypassing interlay.
* Foreign object blocking fluid passage.

2. Too Low NPSH Available

* Suction valve spring too strong.
* Suction valve too heavy.
* Suction line partially clogged.
* Vapor pressure too high.
* Pumping temperature too high.
* Restricted suction pipe fittings

3. Liquid Not Delivered

* Pump not primed.
* Air or vapor pocket in suction line.
* Clogged suction line.
* All suction valves propped open.
* All discharge valves propped open.
* Loose bolts in suction

4. Cavitation

* Low volumetric efficiency.
* Too low NPSH available
* Liquid not delivered.
* Excessive stuffing box leakage.

5. Symptoms Indicating Cavitation

* Stud failure.
* Excessive valve noise.
* Noisy pump operation.

6. **Pump**

* Valve failure.
* Pitting air, stuffing box area, on valves, and plungers
* Overloads Driver
* Pump speed too high.
* Low voltage or other electrical trouble.
* Trouble with engine, turbine gear or other related equipment.
* Excessive discharge line pressure.
* Clogged discharge line.
* Closed or throttled valve in discharge line.
* Incorrect line size for application.
* Improper bypass conditions.
* Over tightened stuffing box glands on adjustable packing.

7. **Stuffing Box Leakage**

* Worn packing.
* Worn rods or plungers.
* Worn stuffing boxes.
* Wrong size packing.

8. **Stud Failure**

* Excessive discharge pressure.
* Improper torquing of nuts.
* Shock overload caused by pump cavitation.

9. **Excessive Valve Noise**

* Broken or weak valve spring.
* Pump cavitation.
* Air leak in suction.

* Air leak in suction piping or loose bolts in pump suction manifold.

10. **Suction or Discharge Line Vibration.**

* Suction line too small.
* Too many bends in suction line.
* Multiple pump installations operating in phase.

11. **Noisy Operation**

(Be sure to differentiate between fluid knock and mechanical knock. Very few knocks are mechanical or new installations.)

* Piston or plunger loose.
* Valve noise amplified through power end.
* Pump cavitation.
* Fluid knock.
* Air leak in suction or loose bolts in pump suction manifold.
* Hydraulic noise in liquid end.
* Loose or worn crosshead pins and bushings.
* Worn connecting rod bearings.
* Worn crossheads.
* Main bearing end play excessive.
* Worn gears.
* Gears out of line.

12. **Broken shafts, bent stripped threads and other catastrophic failures.**

* Start-up against closed gate valve in rods, discharge line.
* Low oil level.
* Contaminated oil.
* Main bearing failure.
* Piston or plunger striking cylinder head.
* Disintegration of worn valves.
* Frozen liquid in fluid body.
* Air leak in suction.
* Loose bolts in pump suction manifold.

13. **Packing Failure**

* Normal wear.
* Improper material.
* Improper lubrication.
* Adjustable packing - gland tightened excessively.

14. **Valve Failure**

* Normal wear.
* Pump cavitation.
* Abrasives in fluid.
* Incompatibility of valve components to corrosive fluid.
* Electrolysis.
* Incorrect installation - Driving on the valve stem, improper torque on jam nut, valve seat and/or valve deck not thoroughly clean an dry.

15. Plunger Failure (Ceramic)

* Thermal shock (cold water hitting hot plunger).
* Packing too tight.
* Suction valve becomes disassembled while in operation.
* Stuffing box gland rubbing on plunger due to improper tightening procedure.

If valve seats are discovered driven too deeply after operation of the pump, look for the following pattern of driven seats indicative of start-up or run against a closed discharge line valve.